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Design of Stop Valves

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DESIGN OF STOP VALVES

By

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PREFACE.

The purpose of this text is to present to engineering students some of the basic principles of machine design and their application to a particular problem.

The stop valve has been selected for illustration as it presents many useful theoretical and practical points in design.

Machine parts not only have to be designed to withstand theoretically the forces acting on them, but they have to be manufactured and fitted in position with the other component parts of the machine. In order to do so, the dimensions obtained theoretically may require to be modified. Where no direct force acts on a part, practical considerations govern the dimensions. It is in this modification. and the fixing of practical sizes that students find most difficulty in design.

To help in these difficulties, suggested proportions are given for some of the valve parts. Also, in the examples worked out to illustrate the method of calculating sizes, modification of such sizes is made where necessary to suit practical considerations.

Professor R. W. Hawken suggests the use of the symbol γ *instead of the usual symbol* y *to denote the distance of the extreme fibre from the neutral axis in a beam cross-section, and this symbol has been used throughout.*

A. R. MUNRO,
14/10/46.

The University of Queensland,
Brisbane.

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Senior Lecturer in Mechanical Engineering, University of Queensland.

Part I.
DESIGN FORMULÆ AND DATA.

The designs presented in this treatise presume that the reader has some knowledge of—

(1) The strength and properties of the materials used for machine parts.

(2) The method of manufacture of machine parts in the workshop.

(3) The method of designing the types of fastenings used in machine construction, e.g., bolts, rivets, keys, pins, cotters, shafts, etc.

(4) The making of working drawings of machine parts.

(5) The loading on machine parts, and factors of safety.

(6) The calculation of bending moments and shearing forces for simple beams.

(7) The method of calculating the moment of inertia, and section modulus for the cross-sectional areas of machine parts acting as loaded beams.

(8) The practical use of a good pocket-book for mechanical engineers.

Important theoretical and practical design data have been compiled by the British Standards Association and the Australian Standards Association and constant reference should be made to the British Standard (B.S.) and Australian Standard (S.A.A.) codes relating to materials, keys, pipe flanges, spur and helical gearing, bevel gearing, worm gearing, etc.

Useful design data are given herewith in a number of data sheets, and although these sheets have been compiled principally for the designs presented, the information given is applicable to any mechanical design.

Machines are a collection of parts taking direct stress or acting as loaded beams.

Forces acting without leverage produce direct stress, and forces acting with leverage produce a moment.

The forces applied from external sources to a machine part have to be resisted by the strength of the fibres of the material of the machine part, and, the moments of external forces require to be balanced by the moments of the internal resistance of the material.

Design, therefore, from the strength standpoint is a balance of internal strength against external force.

The method of design for any machine part comprises :—

(1) A careful analysis of the magnitude and direction of all the forces acting on it, and, whether they produce simple or complex stress.

(2) Its theoretical design so that it will safely withstand the applied forces with the least expenditure of metal.

(3) Its modification so that it can be economically manufactured.

(4) The making of a complete working drawing giving all data as to dimensions, materials, number required, and surfaces that have to be machined.

These four steps in design may therefore be put briefly thus—Analyse, theorise. modify and specify.

Fundamental design formulæ

(1) For forces acting without leverage

The external force = The internal resistance

or $$P = Af$$

where P = the applied load in lb.

A = the cross-sectional area of the machine part in sq. in.

f = the safe allowable working stress for the material in lb. per sq. in.

The stress f may be denoted by f_t if the part is in tension, f_c if in compression, f_s if in shear, and f_b if the part is subjected to crushing or bearing.

(2) For forces acting with leverage

The external moment = The internal moment of resistance

or $$B.M. = f\frac{I}{\gamma}$$

Where $B.M.$ = the bending moment—in. lb.

f = the stress on the extreme fibres—lb. per sq. in.

I = the moment of inertia of the cross-section about its neutral axis—in^4.

γ = the distance from the neutral axis to the extreme fibre of the cross-section—in.

The internal moment of resistance is deduced from the theory of beams and may be written as $f\frac{I}{\gamma}$ or fZ where Z is termed the modulus of the cross-section.

Z therefore equals $\frac{I}{\gamma}$ its units being inches^3.

As a general guide in applying these two fundamental formulæ to practical design the following data sheets are given :—

Data Sheet No. 1 gives the maximum bending moments for a few typical cases of loaded beams.

No. 2 gives the method of arriving at a suitable factor of safety, and values of safe working stresses based on average values for the ultimate strength of the materials given.

No. 3 gives the moment of inertia I and section modulus Z for the following areas—rectangle, square, triangle, circle, semi-circle and ellipse.

No. 4 shows how the position of the neutral axis N.A. for any cross-sectional area can be calculated.

No. 5 shows how the moment of inertia can be calculated.

Further useful data are given on Sheets No. 6, 7 and 8.

Data Sheet No. 6 gives the safe load for M.S. bolts, and the proportions of gear wheel teeth.

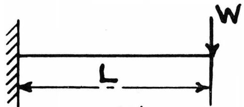
No. 7 gives the method of calculating the strength of gear wheel teeth.

No. 8 gives data for the design of shafting and keys.

BENDING MOMENTS — DATA SHEET No. 1

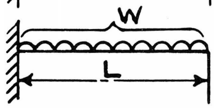
CANTILEVER

CONCENTRATED LOAD



MAX. B.M. $= W \times L$

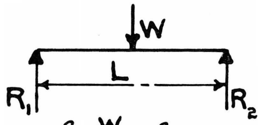
DISTRIBUTED LOAD



MAX. B.M. $= W \times \frac{L}{2}$

SUPPORTED BEAMS

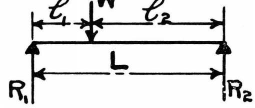
CONCENTRATED LOAD AT CENTRE



$R_1 = R_2 = \frac{W}{2}$

MAX. B.M. $= \frac{W \times L}{4}$

CONCENTRATED LOAD NOT AT CENTRE

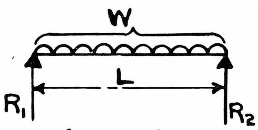


$R_1 = \frac{W \times l_2}{L}$

$R_2 = \frac{W \times l_1}{L}$

MAX. B.M. $= R_1 \times l_1 = \frac{W \times l_2 \times l_1}{L}$

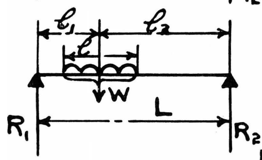
DISTRIBUTED LOAD OVER WHOLE LENGTH



$R_1 = R_2 = \frac{W}{2}$

MAX. B.M. $= \frac{W \times L}{8}$

DISTRIBUTED OVER PORTION OF BEAM



$R_1 = \frac{W \times l_2}{L}$

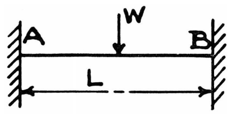
$R_2 = \frac{W \times l_1}{L}$

MAX. B.M. $= R_1 \times l_1 - \frac{W}{2} \times \frac{l}{4}$

$= \frac{W \times l_2 \times l_1}{L} - \frac{W \times l}{8}$

FIXED BEAMS

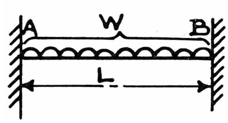
CONCENTRATED LOAD AT CENTRE



MAX. B.M. $= \frac{W \times L}{8}$

THIS IS B.M. AT CENTRE AND AT ENDS A & B

DISTRIBUTED LOAD OVER WHOLE LENGTH

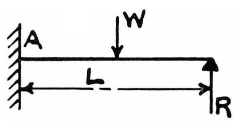


MAX. B.M. AT ENDS A & B $= \frac{W \times L}{12}$

B.M. AT CENTRE $= \frac{W \times L}{24}$

BEAMS FIXED AT ONE END AND SUPPORTED AT OTHER END

CONCENTRATED LOAD AT CENTRE

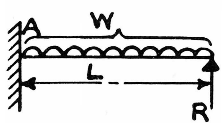


$R = \frac{5}{16} W$

MAX. B.M. AT A $= \frac{3}{16} W \times L$

B.M. AT CENTRE $= \frac{5}{32} W \times L$

DISTRIBUTED LOAD OVER WHOLE LENGTH



$R = \frac{3}{8} W$

MAX. B.M. AT A $= \frac{W \times L}{8}$

DATA SHEET No. 2.

Factors of Safety.

The factor of safety $= a \times b \times c \times d$

where $a = \dfrac{\text{ultimate strength}}{\text{elastic limit}} = \text{say } 2$

$b =$ factor for repetition of load $= 2$ for load in one direction.
$= 3$ for reversal of load.

$c =$ factor for shock $= 1{\cdot}25$ to $1{\cdot}50$

$d =$ factor for uncertainties and unknown contingencies
$= 1{\cdot}5$ to $1{\cdot}75$ for forgings
$= 1{\cdot}75$ to $2{\cdot}25$ for castings

Factors as follows may be taken :—

——	Dead load ($a \times d$).	Live load ($a \times b \times d$) without shock.	
		In one direction.	Reversal.
Forgings	3·3	6·6	10·0
Castings	4·0	8·0	12·0

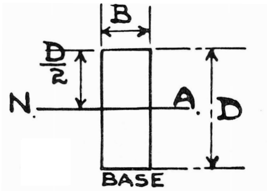
Safe working stresses in lb. per sq. in.

Material.	Ultimate tensile strength.	Dead load.	Live load stress of one kind.	Live load alternate stresses.
Cast iron ..	20,000	5,000 tension 10,000 compression 4,000 shear	2,500 tension 5,000 compression 2,000 shear	1,700 tension and compression
Cast steel (castings)	60,000	15,000 tension 15,000 compression 11,000 shear	7,500 tension 7,500 compression 5,500 shear	5,000 tension and compression
Mild steel (forgings)	70,000	20,000 tension 20,000 compression 15,000 shear	10,000 tension 10,000 compression 7,500 shear	6,600 tension and compression
Gun-metal ..	34,000	8,500 tension 8,500 compression 5,000 shear	4,300 tension 4,300 compression 2,500 shear	2,800 tension and compression
Phosphor bronze	58,000	14,000 tension 14,000 compression 8,000 shear	7,000 tension 7,000 compression 4,000 shear	4,700 tension and compression

DATA SHEET N° 3

MOMENT OF INERTIA I AND SECTION MODULUS Z

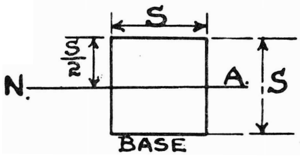
RECTANGLE



$$I_{NA} = \frac{BD^3}{12} \qquad Z = \frac{BD^2}{6}$$

$$I_{BASE} = \frac{BD^3}{3}$$

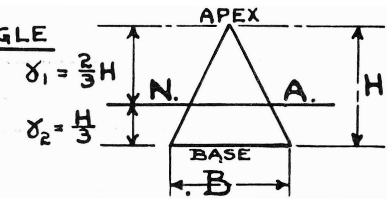
SQUARE



$$I_{N.A.} = \frac{S^4}{12} \qquad Z = \frac{S^3}{6}$$

$$I_{BASE} = \frac{S^4}{3}$$

TRIANGLE



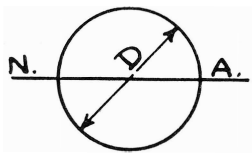
$$I_{NA} = \frac{BH^3}{36} \qquad Z = \frac{I_{NA}}{\gamma_1} = \frac{BH^2}{24}$$

$$Z = \frac{I_{NA}}{\gamma_2} = \frac{BH^2}{12}$$

$$I_{BASE} = \frac{BH^3}{12}$$

$$I_{APEX} = \frac{BH^3}{4}$$

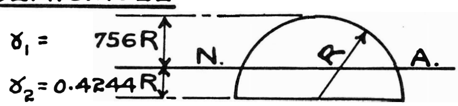
CIRCLE



$$I_{NA} = \frac{\pi D^4}{64} \qquad Z = \frac{\pi D^3}{32}$$

$$I_{POLAR} = \frac{\pi D^4}{32} \qquad Z = \frac{\pi D^3}{16}$$

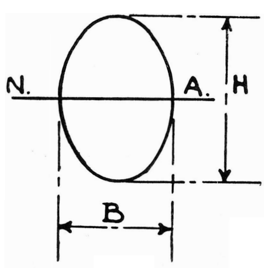
SEMICIRCLE



$$I_{N.A.} = 0{\cdot}1098\,R^4 \qquad Z = \frac{I_{NA}}{\gamma_1} = 0{\cdot}1907\,R^3$$

$$Z = \frac{I_{NA}}{\gamma_2} = 0{\cdot}2586\,R^3$$

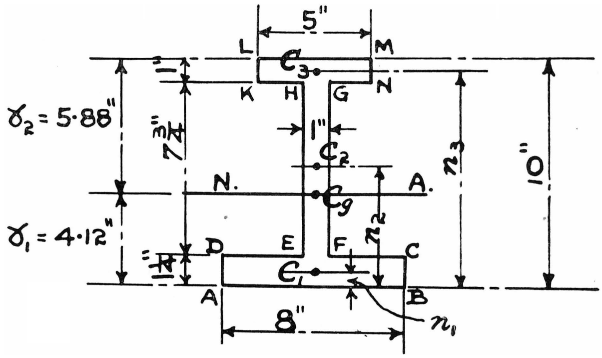
ELLIPSE



$$I_{N.A.} = \frac{\pi}{64} BH^3 \qquad Z = \frac{\pi}{32} BH^2$$

DATA SHEET No. 4.

Calculation for position of N.A. of a cross-sectional area.



The *N.A.* passes through the centroid *Cg* of the area.

The centroid is determined by taking moments of areas about *AB*.

The cross-section is made up of three rectangles, viz., *ABCD, EFGH,* and *KLMN* and the area of each rectangle is taken as concentrated at its own centroid.

The centroid C_1 of area *ABCD* is distant n_1 from *AB*

The centroid C_2 of area *EFGH* is distant n_2 from *AB*.

The centroid C_3 of area *KLMN* is distant n_3 from *AB*.

Let γ_1 be the distance from *AB* to the centroid C_g, then the whole area of the cross-section multiplied by the distance γ_1 equals the sum of the moments of each area about *AB*.

Denoting the areas by A_1 A_2 and A_3 we have

$$\begin{aligned}
\text{Area} \times \text{lever} &= \text{moment} \\
A_1 \times \quad n_1 &= A_1 n_1 \\
A_2 \times \quad n_2 &= A_2 n_2 \\
A_3 \times \quad n_3 &= A_3 n_3 \\
\text{Whole Area} \times \quad \gamma_1 &= \text{Total moment} \\
\textit{i.e.,}\ (A_1 + A_2 + A_3) \times \quad \gamma_1 &= A_1 n_1 + A_2 n_2 + A_3 n_3
\end{aligned}$$

The calculation may be set out as follows :—

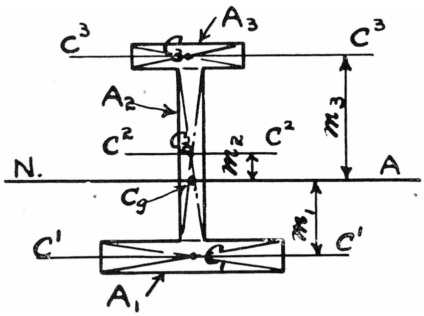
Sq. in.	Area	×	Lever	=	moment		
10·00	$8'' \times 1\frac{1}{4}''$	×	$\frac{5}{8}''$	=	6·25	rectangle	*ABCD*
7·75	$7\frac{3}{4}'' \times 1''$	×	$5\frac{1}{8}''$	=	39·719	,,	*EFGH*
5·00	$5'' \times 1''$	×	$9\frac{1}{2}''$	=	47·50	,,	*KLMN*
22·75		×	γ_1	=	93·469		

$$\therefore \gamma_1 = \frac{\text{Total moment}}{\text{whole area}} = \frac{93{\cdot}469}{22{\cdot}75} = 4{\cdot}12''$$

and $\gamma_2 = 10'' - 4{\cdot}12'' = 5{\cdot}88''$.

DATA SHEET No. 5.

Calculation of Moment of Inertia I of a cross-sectional area about its N.A.



Refer to Data Sheet No. 4 for dimensions.

Taking axes parallel to $N.A.$ through C_1 C_2 and C_3

then for rectangle A_1 — $I_{NA} = I_{C1C1} + A_1 m_1^2$

for rectangle A_2 $\quad I_{NA} = I_{C2C2} + A_2 m_2^2$

for rectangle A_3 $\quad I_{NA} = I_{C3C3} + A_3 m_3^2$

and the moment of inertia of the whole cross-section is the sum of these three moments of inertia.

From data sheet No. 3 $I_{NA} = \dfrac{bd^3}{12}$ for a rectangle

$$\therefore I_{C1C1} = \frac{8 \times 1\frac{1}{4}^3}{12} = 1{\cdot}28 \text{ in.}^4 \text{ for area } A_1$$

$$I_{C2C2} = \frac{1 \times 7\frac{3}{4}^3}{12} = 38{\cdot}75 \text{ in.}^4 \text{ for area } A_2$$

$$I_{C3C3} = \frac{5 \times 1^3}{12} = 0.417 \text{ in.}^4 \text{ for area } A_3$$

$m_1 = 3{\cdot}5''$, $m_2 = 1{\cdot}06''$ and $m_3 = 5{\cdot}38''$.

$$\therefore I_{NA} = (1{\cdot}28 + 10 \times 3{\cdot}5^2) + (38{\cdot}75 + 7{\cdot}75 \times 1{\cdot}06^2) + (0{\cdot}417 + 5 \times 5{\cdot}38^2)$$
$$= 123{\cdot}78 \quad + 47{\cdot}43 \quad + 145{\cdot}12$$
$$= 316{\cdot}33 \text{ in}^4.$$

The complete calculation incorporating Data Sheet No. 4 may be set out in tabulated form as follows :—

Sq. in.	A	n	An	I_{CC}	m	m^2	Am^2
10·00	$8 \times 1\frac{1}{4}$	$\frac{5}{8}$	6·25	1·28	3·5	12·25	122·5
7·75	$7\frac{3}{4} \times 1$	$5\frac{1}{8}$	39·719	38·75	1·06	1·123	8·68
5·00	5×1	$9\frac{1}{2}$	47·50	0·417	5·38	28·94	144·7
ΣA 22·75			ΣAn 93·469	ΣI_{CC} 40·447			ΣAm^2 275·88

$$\gamma_1 = \frac{\Sigma An}{\Sigma A} = \frac{93{\cdot}469}{22{\cdot}75} = 4{\cdot}12''$$

$$I_{NA} = \Sigma I_{CC} + \Sigma Am^2 = 40{\cdot}477 + 275{\cdot}88 = 316{\cdot}33 \text{ in.}^4.$$

DATA SHEET No. 6.

Bolts—

Safe loads for M.S. bolts—(*a*) for connecting machine parts subject to a live load, and (*b*) suitable for fluid tight joints :—

Diameter in inches.	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2
*Safe load in lb. . .	250	500	1,000	1,500	2,400	3,500	4,800	6,300	8,000	12,500	16,500

* Based on an average curve obtained by plotting the safe loads given in—
Seaton and Rounthwaite's Pocket Book.
Principles of Machine Design.—Norman.
Machine Design.—Hyland & Kommers.

For a dead load these values may be increased by 50 per cent.

For shock half of these values can be taken.

Gear wheel teeth (Involute form)—

$$\text{addendum} = \text{module} = \frac{1}{\text{diametral pitch}}$$

$$\text{Dedendum} = \text{addendum} + \frac{0{\cdot}157}{\text{diametral pitch}}$$

$$\text{Diametral pitch} = \frac{\pi}{\text{circular pitch}}$$

Diametral pitch D.P.	Circular pitch C.P.	Thickness.	Addendum.	Dedendum.
1	3·142	1·571	1·000	1·157
$1\frac{1}{4}$	2·513	1·257	0·800	0·926
$1\frac{1}{2}$	2·094	1·047	0·667	0·771
$1\frac{3}{4}$	1·795	0·898	0·571	0·661
2	1·571	0·785	0·500	0·578
$2\frac{1}{4}$	1·396	0·698	0·444	0·514
$2\frac{1}{2}$	1·257	0·628	0·400	0·463
$2\frac{3}{4}$	1·142	0·571	0·364	0·421
3	1·047	0·524	0·333	0·386
$3\frac{1}{2}$	0·898	0·449	0·286	0·331
4	0·785	0·393	0·250	0·289
5	0·628	0·314	0·200	0·231
6	0·524	0·262	0·167	0·193
7	0·449	0·224	0·143	0·165
8	0·393	0·196	0·125	0·145
9	0·349	0·176	0·111	0·129
10	0·314	0·157	0·100	0·116

DATA SHEET No. 7.

Design of gear wheel teeth.

Lewis formula

$$P = f \times b \times p \times y$$

where P = load at pitch circle in lb.

f = safe working stress in lb. per sq. in.

b = width of wheel rim in inches

p = circular pitch in inches

y = tooth factor depending on number of teeth in wheel

For 20° involute teeth

$$y = 0{\cdot}154 - \frac{0{\cdot}912}{n} \text{ where } n = \text{number of teeth.}$$

*safe working stress $f = C\left(\dfrac{600}{600 + V}\right)$

where V = velocity of rim in ft. per min.

and C = 8,000 for cast iron

= 12,000 for bronze

= 20,000 for steel castings

= 25,000 for mild steel

= 80,000 for heat treated nickel and chrome steels

Teeth can be proportioned for strength by the above method.

If wear is a vital consideration the size selected can be checked from formula and charts given in the British Standards, or from formula and tables given in pocket-books.

Refer to—

British Standard No. 436—Spur and helical gearing.

British Standard No. 545—Bevel gearing.

British Standard No. 721—Worm gearing.

Pocket-book for Mechanical Engineers by D. A. Low.

Mechanical World Year Book.

* Machine Design.—Albert.

DATA SHEET No. 8.

Design of Shafts.

Twisting only

$$\text{T.M.} = f_s \frac{I_{\text{polar}}}{y} = f_s \frac{\frac{\pi d^4}{32}}{\frac{d}{2}} = \frac{\pi d^3}{16} f_s$$

$$= 0{\cdot}196\, d^3 f_s$$

where d = diameter of shaft in inches

f_s = 8,000 lb. per sq. in. for M.S.

Twisting and Bending

Equivalent T.M. = $\sqrt{BM^2 + TM^2}$. Guest formula

and Equiv. $T.M. = 0{\cdot}196\, d^3 f_s$

where $f_s = \frac{f_t}{2} = \frac{10{,}000}{2} = 5{,}000$ lb. per sq. in for M.S.

Allowance for keyways

S.A.A. code No. CB2 (crane and hoist code)

$$\text{gives } K = \left(1 - \frac{0{\cdot}2b}{D} - \frac{1{\cdot}1d}{D}\right)$$

where K = ratio of strength of keywayed shaft to strength of uncut shaft

b = breadth of keyway in inches

d = depth of keyway in inches

D = diameter of shaft in inches.

Keys

Proportions may be taken as follows or obtained from British Standards No. 46.

Breadth $= \frac{\text{shaft diameter}}{4} + \frac{1}{8}$ in.

Thickness $= \frac{3}{8}$ breadth $+ \frac{1}{8}$ in.

Taper $= \frac{1}{8}$ in. per foot length

Gib heads — height = 1·7 thickness

length = 2·0 thickness

Design

Keys are designed for shear

$$P = Af_s = B \times L \times f_s$$

$$\therefore L = \frac{P}{B \times f_s}$$

where P = load on key in lb.

L = length of key in inches

B = breadth of key in inches

f_s = safe working stress in lb. per sq. in. = 8,000 for M.S.

Part II.

THE DESIGN OF STEAM STOP VALVES WHICH ARE OPERATED BY A HANDWHEEL ATTACHED TO THE SPINDLE.

There are two types of stop valve bodies, viz., the right angle pattern and the globe pattern. Fig. 1 shows a sectional elevation of a right angle pattern valve and fig. 2 of a globe pattern valve. The complete valve assembly comprises the following details :—

1. The main body which may be made of cast iron or gun-metal for low pressures and of cast steel for high pressures.*
2. The valve seating bush made of bronze.
3. The valve made of bronze.
4. The valve spindle made of steel, muntz metal or bronze.
5. The cover made of the same material as the main body.
6. The stuffing box which is cast with the cover, the neck bush and gland being made of gun-metal.
7. The bridge columns made of mild steel.
8. The bridge or crosshead made of mild steel or cast iron and having a gun-metal screwed bush for the spindle.

Note in fig. 1 the bridge and columns are cast with the cover.

9. The operating hand wheel usually made of cast iron.

The valve body is arranged so that the valve closes against the pressure, which would allow the stuffing box to be repacked, or an additional turn of packing to be added when the valve is closed.

The valve is attached loosely to the spindle so as to enable it to accommodate itself to its seat. A usual method of attachment is to make the top of the valve in the form of a horseshoe into which the bottom of the spindle fits. Due to this loose attachment it is necessary to guide the valve on to its seat. The guiding is effectively done by constructing the valve with three or four webs as in the valve shown in fig. 1, or the valve may have a central stem which moves in a guide provided in the valve bush as shown in the globe pattern valve in fig. 2.

---

* British Standards specify that—

(*a*) Cast iron shall not be used for steam pressures exceeding 150 lb. per sq. in. (guage), superheated steam at boiler pressure, or temperatures above 400° Fah.

(*b*) Bronze shall not be used for superheated steam at boiler pressure, or temperatures above 425° Fah.

Valve seating bush.

Fig. 3 shows a gun-metal bush suitable for low pressure valves up to about $3\frac{1}{2}$ in. diameter. These are machined all over and driven tightly into the main casting. Due to unequal expansion and contraction of valve bush and body casting it is advisable in larger diameter valves to secure the bush rigidly by increasing the width of the flange and screwing it down with three or four set screws or studs as in fig. 4. An alternative method is to screw the bush itself into the body casting as illustrated in fig. 5. When the body casting is made of gun-metal the bush is sometimes dispensed with, but as it is a relatively small detail and easily renewed it would be good practice to fit one in gun-metal construction.

Diameter of bush.

The clear area through the bush must be equal to the cross-sectional area of the steam pipe. The obstruction caused by the guide webs in the valve can be taken as approximately 15 per cent. of the pipe area

$$\therefore \text{ area through bush} = 1{\cdot}15 \text{ pipe area}$$

$$\text{or diameter of bush } d_1 = \sqrt{1{\cdot}15\, \mathrm{d}^2}$$

where d = diameter of pipe.

Dimensions of bush.

The thickness t is purely a practical consideration. If made too thin the bush would be liable to burst when being driven into place.

The following proportions are suggested :—

(*Dimensions in inches.*)

Diameter of valve d	2	3	4	5	6	7	8
Thickness of bush t	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$
Thickness of flange T	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$
Width of flange W	$\frac{1}{8}$	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{1}{4}$	$\frac{9}{32}$	$\frac{5}{16}$

minimum dimensions for bushes screwed into the body.

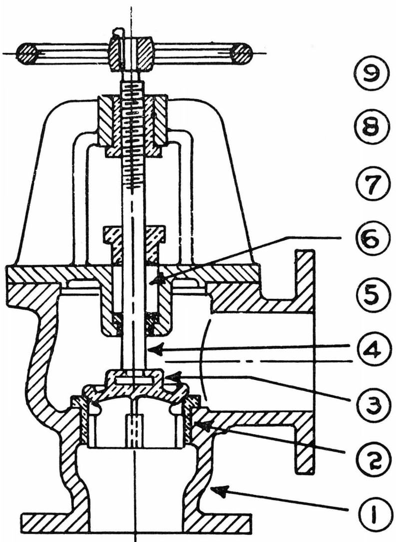


FIG. 1

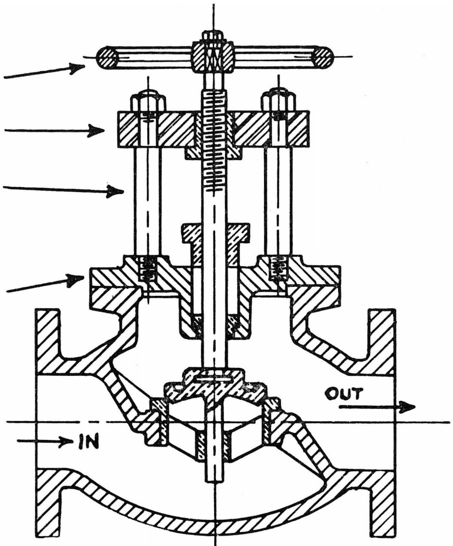


FIG. 2

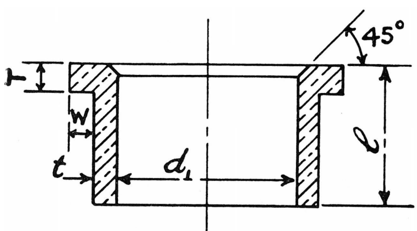


FIG. 3

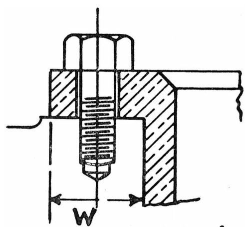


FIG. 4

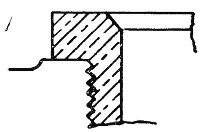


FIG. 5

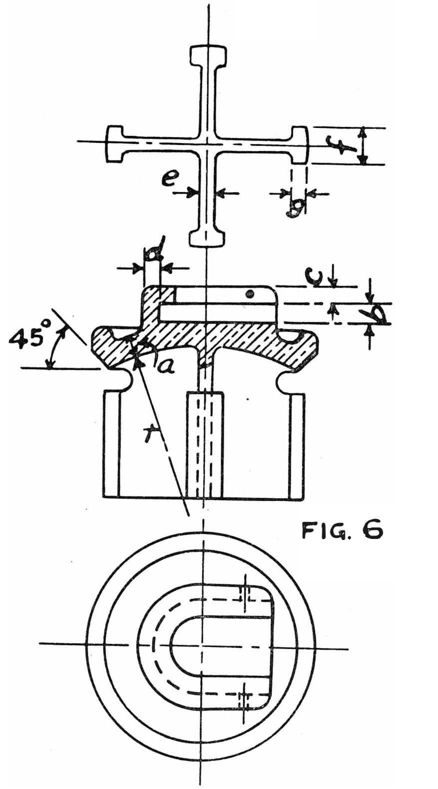


FIG. 6

For bushes fixed with set screws or studs the flange width W will depend on the diameter of the screws and will be at least equal to twice the diameter of the screw plus $\frac{1}{8}$ in., as the heads of the screws must be clear of the valve.

The following is suggested for number and diameter of screws :—

Diameter of valve.	Steam pressure in lb. per sq. in.			
	100	200	300	400
4″	3—$\frac{3}{8}$″	4—$\frac{3}{8}$″	3—$\frac{7}{16}$″	4—$\frac{7}{16}$″
5″	4—$\frac{3}{8}$″	4—$\frac{7}{16}$″	4—$\frac{1}{2}$″	4—$\frac{9}{16}$″
6″	4—$\frac{7}{16}$″	4—$\frac{1}{2}$″	4—$\frac{9}{16}$″	4—$\frac{5}{8}$″
7″	4—$\frac{1}{2}$″	4—$\frac{9}{16}$″	4—$\frac{5}{8}$″	6—$\frac{5}{8}$″
8″	4—$\frac{9}{16}$″	4—$\frac{5}{8}$″	6—$\frac{5}{8}$″	8—$\frac{5}{8}$″

The depth l should be such that when the valve is fully opened the guide webs on the valve are halfway in the bush in order to prevent the valve from canting, as the enlarged ends of the webs are not a tight fit in the bush. The webs should project $\frac{1}{8}$ in. through the bush when the valve is closed.

$$\therefore\ l = \text{twice the valve lift} - \tfrac{1}{8}\text{ in.}$$

Lift of valve.

In the case of a valve without guide webs the bush diameter equals the pipe diameter, and when the valve is opened the area through which the steam flows will be the circumference of the bush multiplied by the lift of the valve, hence circumference of bush $\times$ lift $=$ area of pipe

$$\textit{i.e.}\ \pi d \times \text{lift} = \frac{\pi d^2}{4}$$

$$\therefore\ \text{lift} = \frac{d}{4}$$

The minimum lift should therefore be $\frac{1}{4}$ of the diameter of the bush but to ensure freedom for the passage of the steam it is advisable to make it from 10 per cent. to 20 per cent. greater.

Valve seat.

The angle of the valve seat is made 45° and the width of the seat should be such that the bearing pressure is about 2,000 lb. per sq. in. The seat should be kept as narrow as possible as it is difficult to keep valves tight with wide seatings. Successive re-grindings of the valve widen the seat and lessen the bearing pressure.

Valve.

The sketches in fig. 6 show a typical valve which has four guide webs, and the usual horseshoe construction on the top to accommodate the valve spindle. The bottom of the valve spindle is enlarged in diameter and this enlarged part fits into the recess in the horseshoe. After the valve is slipped on to the bottom of the spindle a split pin is inserted to prevent it from sliding off. The top of the valve may be flat or curved and, in the case of small valves, practical rather then theoretical considerations determine the thickness. The top forms a circular flat plate supported all round the seating and loaded with the load applied to close the valve, this load being distributed over the area of contact of the valve spindle with the valve. The additional metal on the top forming the horseshoe, together with the webs underneath adds considerably to the strength of the plate.

For the condition of a circular flat plate the Mechanical World Year Book gives the following formula deduced from Grashof's maximum stress theory—

$$f = \frac{P}{\pi t^2}\left(2 \log_e \frac{d}{d_1} + \frac{3}{2}\right)$$

where f = stress in lb. per sq. in.

P = load on the spindle (lb.)

t = thickness in inches

d = diameter of valve seating bush in inches.

d_1 = diameter of bottom of spindle in inches.

Thicknesses calculated by this formula would make the valves far too heavy and the following proportions are suggested as suitable :—

Proportions for valves (dimensions in inches).

Diameter of valve.	a	b	c	d	e	f	g	Number of guide webs.	Diameter of split pin.
2	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{5}{32}$	$\frac{3}{8}$	$\frac{3}{16}$	3	$\frac{1}{8}$
3	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{5}{16}$	$\frac{7}{32}$	$\frac{7}{16}$	$\frac{1}{4}$	3	$\frac{5}{32}$
4	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{2}$	$\frac{9}{32}$	4	$\frac{3}{16}$
5	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{5}{16}$	$\frac{9}{16}$	$\frac{11}{32}$	4	$\frac{7}{32}$
6	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{5}{8}$	$\frac{13}{32}$	4	$\frac{7}{32}$
7	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{9}{16}$	$\frac{13}{32}$	$\frac{11}{16}$	$\frac{7}{16}$	4	$\frac{1}{4}$
8	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{3}{4}$	$\frac{15}{32}$	4	$\frac{1}{4}$

radius r = twice the diameter of valve.

For flat valves the dimension a may be increased by $\frac{1}{16}$ in. to $\frac{1}{8}$ in.

A flat valve with a stem is shown in fig. 7. The guide for the stem is provided in the valve seating bush and is an integral part of the bush. If a longer guiding surface is desired the guide can be made as shown on the valve in fig. 2.

Suitable proportions for stem and bush guide are—

$$\text{Diameter of stem} = \frac{\text{diameter of valve}}{8} + (\tfrac{1}{8}\text{ in. to }\tfrac{3}{8}\text{ in.})$$

$$\text{Dimension } h = 2 \times \text{diameter of stem (minimum)}$$

$$\text{Dimension } k = \text{dimension } e \text{ given for webs in fig. 6}$$

$$\text{Dimension } l = \text{diameter of stem } + 1{\cdot}75k.$$

The valve should seat on the bush as shown in fig. 8 such that about $\frac{1}{4}$ face is inside the bush. The thickness of the valve round the seating is increased to counteract the thinning of the metal due to successive grinding. The top of this increased thickness should not be higher than the bottom of the slot in the horseshoe so that the spindle can be slipped in. The valve webs should be well cut away to allow for the machining of the seat. Should it be desired to lessen the weight of the valve, the webs can be cut away as indicated by the dotted curved line.

Spindle.

The spindle is subject to compressive stress and, if fairly long between the bridge and the support given by the neck bush in the stuffing box, it will require to be designed as a column. It may be screwed with either a Whitworth vee thread or a square thread. The length of the screwed portion will require to be equal to the depth of the bridge, plus the valve lift, plus three to five threads.

In order that the valve can be tightly closed the load applied to the spindle per medium of the handwheel should be greater than the load underneath the valve.

Spindles may be designed for the following loads where

$$P = \text{area of valve} \times \text{pressure in lb. per sq. in.}$$

Diameter of valve in inches.	2	3	4	5	6	7	8	9 and larger
Load ..	$2P$	$1{\cdot}7P$	$1{\cdot}5P$	$1{\cdot}4P$	$1{\cdot}35P$	$1{\cdot}3P$	$1{\cdot}25P$	$1{\cdot}2P$

$$\text{Area in sq. in. at bottom of threads} = \frac{\text{Load in lb.}}{\text{safe working stress in lb. per sq. in.}}$$

The value of the safe working stress for the material will be that for a live load producing stress of one kind only.

The spindle is assembled in place by pushing it up through the stuffing box and thus screwing it into the bridge. The dimension for the top end of the spindle will therefore be less than the smallest diameter of the thread in the bridge bush, hence, in settling on a suitable diameter for the spindle the torque applied by the handwheel has to be considered. This point is dealt with under handwheel design.

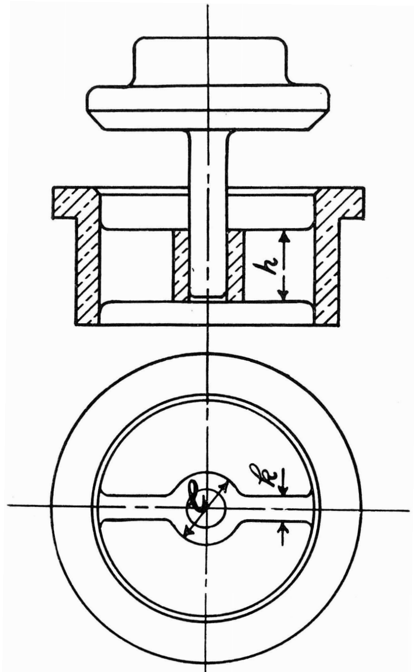


FIG. 7

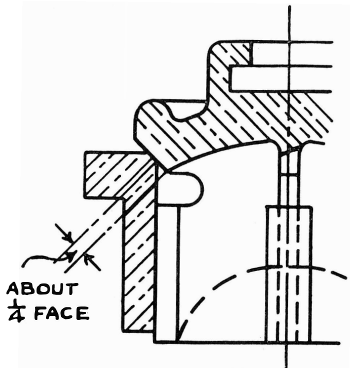


FIG. 8

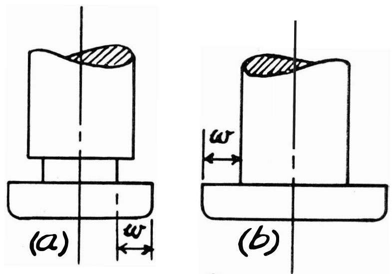


FIG. 9

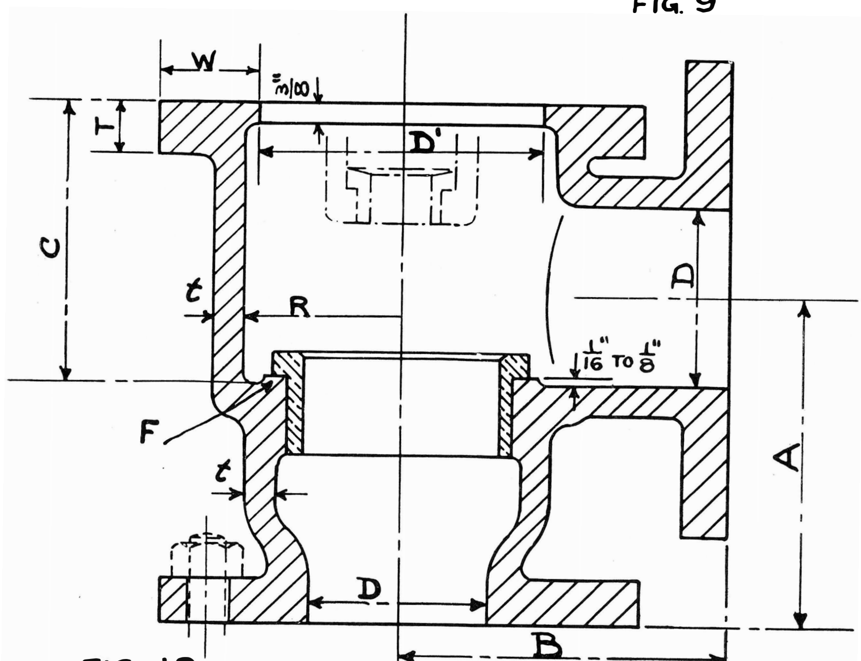


FIG. 10

The bottom of the spindle may be made as shown at (*a*) or (*b*) in fig. 9. The diameter of the recessed portion in (*a*) which fits into the horseshoe on the top of the valve should not be smaller than the diameter at the bottom of the threads.

Suitable dimensions for the width *w* are as follows :—

Diameter of valve—inches.	2	3	4	5	6	7	8	9
Width *w*—inches ..	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{1}{4}$	$\frac{9}{32}$	$\frac{5}{16}$	$\frac{11}{32}$	$\frac{3}{8}$	$\frac{13}{32}$

Body casting for right angle pattern valves.

A sectional elevation of a typical body casting is shown in fig. 10. The machined facing *F* for the seating bush is the natural starting point from which to begin the drawing of the casting. The depth of the facing can be $\frac{1}{16}$ in. or $\frac{1}{8}$ in. and the diameter made $\frac{1}{8}$ in. to $\frac{1}{4}$ in. greater than the diameter of the flange on the valve seating bush. The bottom of the outlet branch is drawn below the facing *F* as indicated. The radius *R* of the body will require to be $\frac{1}{4}$ in. greater than the radius of the facing *F* to provide the two necessary fillets required for the practical manufacture. The radius *R* has also to be sufficient for the flow of the steam past the valve guide webs, and above the valve. If a valve having four guide webs is placed in the position shown in fig. 11 then half the volume of the steam will flow direct to the outlet branch. The other half volume has to get to the outlet branch through *ab* and *cd*, which area for the flow is shown cross-hatched. An inspection of this area shows that between the planes marked *A* and *B* the steam flows direct to the outlet. The remainder of the half volume has to pass upwards above the plane *B* through the area between the two semi-circles of radii *R* and r_1. Suppose $\frac{1}{3}$ of the half volume passes direct to the outlet between the planes *A* and *B*, then $\frac{2}{3}$ of the half volume or $\frac{1}{3}$ of the whole volume has to pass through the semi-circular area.

If r = radius of pipe, and we assume that the radius r_1 of the valve top = 1·1 radius of pipe

we have—

$$\text{semi-circular area} = \frac{\text{area of pipe}}{3}$$

$$i.e. \frac{\pi}{2}(R^2 - 1{\cdot}21r^2) = \frac{\pi}{3}r^2$$

$$\frac{\pi}{2}R^2 = \frac{\pi}{3}r^2 + \frac{\pi}{2} \times 1{\cdot}21\, r^2$$

$$\frac{R^2}{2} = \frac{r^2}{3} + 0{\cdot}605\, r^2$$

$$= 0{\cdot}938 r^2$$

$$R^2 = 1{\cdot}976 r^2 = \text{say } 2r^2$$

$$\therefore R = \sqrt{2r^2} = 1{\cdot}4\, r.$$

The radius R may therefore be assumed by drawing it from the practical consideration or making it 1·4 times the radius of the pipe—whichever is the greater. From the dimensions of the valve, seating bush, valve lift and height to bottom of cover a check can be made, if necessary, to see that the area for flow is sufficient.

Thickness of body metal.

The thickness t is calculated from the pipe formula and a practical constant added to cover porosity in the metal and shift of core during manufacture.

The practical constant may be taken as 0·35 in. to 0·4 in. for cast iron and 0·25 in. to 0·3 in. for bronze or cast steel.

thus $t = \dfrac{pd}{2f} + \text{constant}$

where t = thickness of metal in inches

p = pressure in lb. per sq. in.

d = diameter of pipe in inches.

f = safe working stress in lb. per sq. in. = 2,500 lb. per sq. in. for cast iron, 4,300 lb. per sq. in. for gun-metal, and 7,500 lb. per sq. in. for cast steel, for materials with ultimate strength as given on data sheet No. 2.

The thickness of the metal below the bush and at the outlet branch could be less since the diameters are smaller, but it is best to keep it uniform and to calculate it from the larger radius.

The body outline given in fig. 10 allows for the over-running of the tool in machining the bore for the bush. This is a good practical point, for, if the bush is fitted as in fig. 12 there would be a tendency for the tool to dig in at the corner M and thus weaken the casting ; also, if the bush flange is to bed properly on the facing F a small clearance between the bottom of the bush and the body casting is necessary.

The flanges at the inlet and outlet branches are proportioned according to the British Standards.

The dimension A should be sufficient to allow for easy radii from the enlarged diameter to the pipe diameter D, and must permit of putting the nut on the stud or bolt.

The dimension B is governed mainly by the diameter of the cover and by being able to bolt the pipe to the outlet branch.

The height C depends on the lift of the valve and the proportions of the stuffing box.

The opening D^1 in the cover flange must be greater than the diameter of the flange on the valve seating bush, otherwise it would be impossible to fit the bush in position. This opening is machined so that the cover may fit exactly into it in order to keep the spindle central with the valve bush.

Top flange on body for cover.

The joint between the cover and the body casting has to be steam tight, in which case the width of the joint W should not be less than 2·8 times the diameter of the stud fitted. The thickness of the flange T should be 1·5 times the stud diameter for cast iron, and not less than the stud diameter for gun-metal or cast steel.

The number and diameter of the studs required are determined from the maximum load and the allowable pitch necessary for steam tightness.

Since the bridge is integral with, or attached to the cover, the whole of the spindle thrust comes on to the cover studs when the valve is closed.

If the inlet and outlet directions are reversed, the pressure is above the valve when the valve is closed, and in the worst case we have—

Maximum load on studs = area of opening D^1 × steam pressure + spindle thrust.

Diameter of cover studs.

To satisfy load

It is usual to select an even number of studs thus the load on each stud =

$\dfrac{\text{maximum load}}{n}$ where n = number of studs.

The required diameter can be obtained from data sheet No. 6.

To satisfy steam tightness.

The pitch may vary from 3·5 to 6·0 times the diameter of the stud according to the pressure. For low pressures the value of 6·0 times should not be exceeded. To fit a spanner on the nuts a minimum pitch of 3·0 times the stud diameter would be required.

The diameter of the pitch circle for the studs

$= D^1 + 2{\cdot}8$ times stud diameter.

or $= D^1 + 2\ \{W - (\text{stud diameter} + \tfrac{1}{4}'')\}$

The following example will show the method of determining a suitable diameter of stud.

Given—Valve opening $D^1 = 9\tfrac{1}{2}$ in. diameter

Steam pressure = 400 lb. per sq. in.

Spindle thrust = 16,000 lb.

Load on studs—

Due to steam pressure $=$ area $9\frac{1}{2}''$ dia. $\times$ 400

$= 28{,}000$ lb.

Thrust from spindle in final closing of valve $= 16{,}000$ lb.

Total load $= 44{,}000$ lb.

From data sheet No. 6

Safe load for $1\frac{1}{8}$ in. dia. stud $= 3{,}500$ lb.

Safe load for $1\frac{1}{4}$ in. dia. stud $= 4{,}800$ lb.

Safe load for $1\frac{3}{8}$ in. dia. stud $= 6{,}300$ lb.

thus to satisfy the load either 14—$1\frac{1}{8}$ in. dia., or 10—$1\frac{1}{4}$ in. dia. or 8—$1\frac{3}{8}$ in. dia. studs would be required.

An even number of studs is taken for the purpose of accommodating the column portions of the bridge.

Taking width of joint $= 2{\cdot}8 \times$ dia. of stud.

for $1\frac{1}{8}$ in. dia. studs width $W = 2{\cdot}8 \times 1{\cdot}125 = 3\frac{1}{8}$ in.

$1\frac{1}{4}$ in. dia. studs width $W = 2{\cdot}8 \times 1{\cdot}25 = 3\frac{1}{2}$ in.

$1\frac{3}{8}$ in. dia. studs width $W = 2{\cdot}8 \times 1{\cdot}375 = 3\frac{7}{8}$ in.

then diameter of pitch circle $= D^1 + 2 \left\{W - (d + \frac{1}{4}\text{ in.})\right\}$

for $1\frac{1}{8}$ in. studs $= 9\frac{1}{2}'' + 2\,(3\frac{1}{8}'' - 1\frac{3}{8}'') = 13$ in.

$1\frac{1}{4}$ in. studs $= 9\frac{1}{2}'' + 2\,(3\frac{1}{2}'' - 1\frac{1}{2}'') = 13\frac{1}{2}$ in.

$1\frac{3}{8}$ in. studs $= 9\frac{1}{2}'' + 2\,(3\frac{7}{8}'' - 1\frac{5}{8}'') = 14$ in.

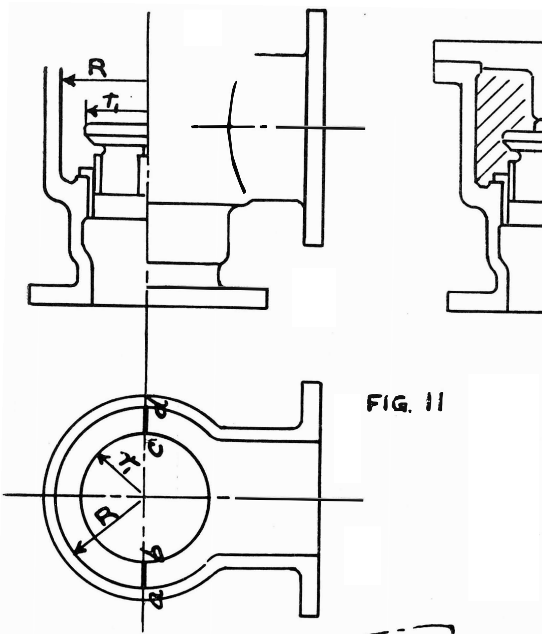
$$\text{Pitch of studs} = \frac{\text{circumference of pitch circle}}{\text{number of studs.}}$$

for $1\frac{1}{8}''$ dia. $= \dfrac{40{\cdot}84}{14} = 2{\cdot}92$ in. *i.e.* $2{\cdot}59 \times$ dia. of stud

$1\frac{1}{4}''$ dia. $= \dfrac{42{\cdot}4}{10} = 4{\cdot}24$ in. *i.e.* $3{\cdot}39 \times$ dia. of stud

$1\frac{3}{8}''$ dia. $= \dfrac{43{\cdot}98}{8} = 5{\cdot}5$ in. *i.e.* $4{\cdot}0 \times$ dia. of stud

The $1\frac{1}{8}$ in. dia. studs give a pitch which is below the minimum of 3 times stud diameter and for the high pressure considered 4 times stud diameter may be on the large side therefore use 10—$1\frac{1}{4}$ in. diameter.



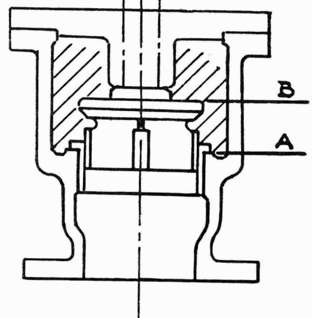


FIG. 11

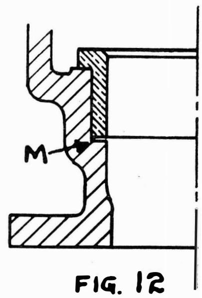


FIG. 12

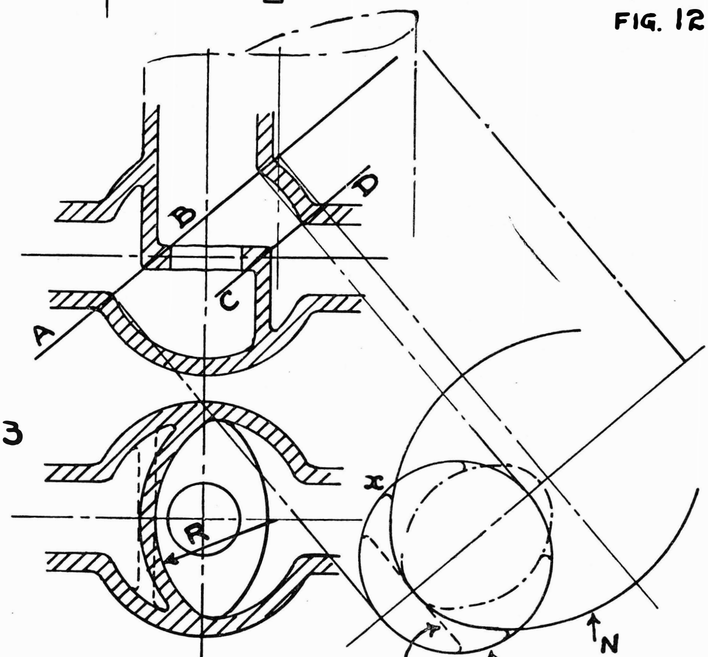


FIG. 13

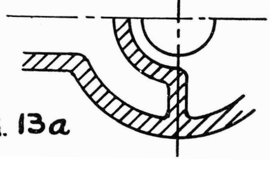


FIG. 13a

Body casting for Globe pattern valves.

The body casting for small diameter valves may be made spherical shaped and the partitioning metal taken parallel to the spindle as in fig. 13. With the larger valves it is necessary to make the body barrel shaped and to incline the partition metal as shown in fig. 2 in order to give an easy flow to the steam from the pipe to the valve.

The area of the passage at the sections AB and CD must be equal to the area of the pipe and should preferably be greater to allow for any errors in shape during manufacture. A greater excess is allowed in small valves than in large valves and may be taken as follows :—

Diameter of pipe.	1 in.	3 in.	6 in.	9 in.
Percentage excess area	60	30	20	10

The partition metal may run straight across the body but is usually curved as when curved it gives greater cross-sectional areas at AB and CD.

When the partition metal is curved with a radius R the projection of the cross sectional area through AB is obtained by the circle M giving the section of the sphere, and the ellipse N for the cylinder of radius R. If the partition metal runs straight across as indicated by the dotted lines the area is given by the circle M and the dotted straight line Q.

It can be seen that the area through the section AB will increase as the radius R is decreased. Decreasing R to its smallest dimension for accommodating the valve or valve seating bush would result in the partitioning metal being shaped as in fig. 13*a*.

The intersections of the circle and ellipse are rounded as at x to allow for the filleting of the partition metal into the body.

The cross-section through barrel shaped bodies is constructed as shown in fig. 14. Since all cross-sections parallel to the ends are circular, a series of planes a, b, c, d . . etc., are taken through the barrel and the common points obtained where each of these planes and the section plane AB intersects the surface. For example, taking plane b the common point in the elevation is b_1 which projected on to the circle representing plane b gives the two points $b_2 b_2$. Projecting perpendicular to AB from b_1 the distance $b_3\ b_3$ is set off equal to the distance $b_2 b_2$. The widths for the planes a, c, d, . . . etc., are obtained in a similar manner by projecting from $a_1\ c_1\ d_1$ . . . etc.

The horizontal portion of the partition metal has to be modified to suit the valve seating bush. On the top side a machined facing is necessary, and on the underside a cylindrical boss of sufficient length for the valve bush, thus the central portion of the body can be made as shown in fig. 15. An analysis of this modified condition shows that the cross-section at AB is influenced by the cylinder of diameter C, the

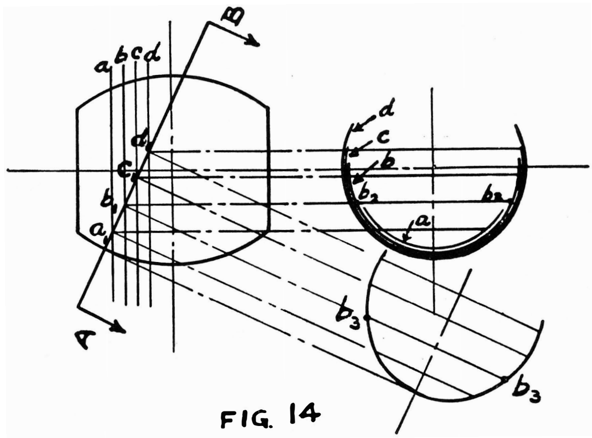


FIG. 14

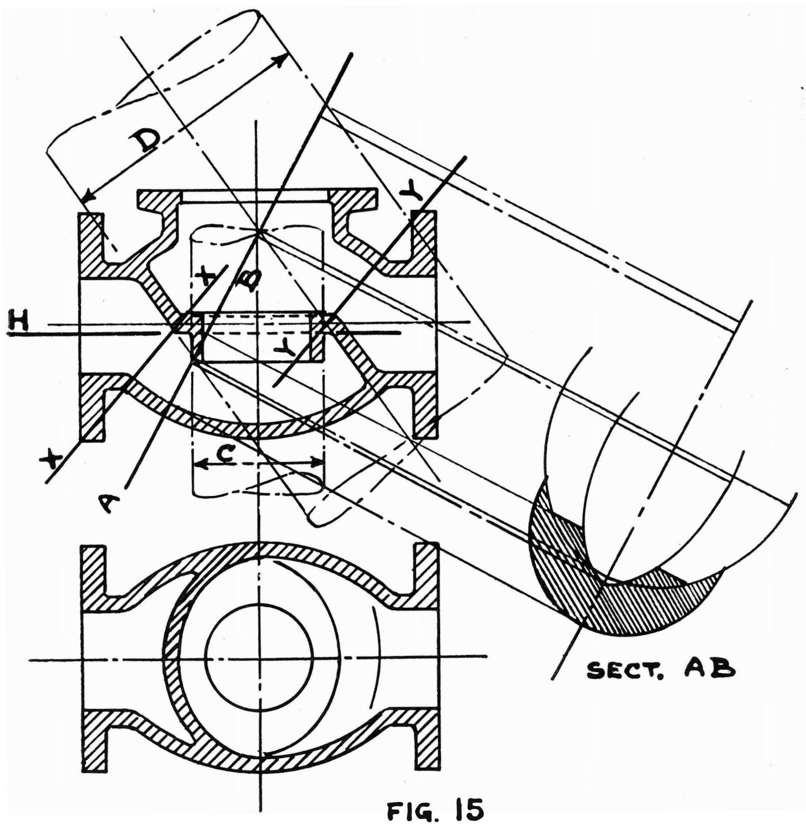


FIG. 15

horizontal plane H, and the cylinder of diameter D, resulting in the cross-sectional area shown in the diagram. The cross-sections at XX or YY are influenced only by the cylinder of diameter D. In all cases of design it is advisable to make a preliminary drawing of the body casting with the valve bush in position and to obtain the areas at such cross-sections as AB, XX, and YY.

The following example will show the method of designing the body casting for a Globe pattern valve :—

Diameter of pipe or valve	6 in.
Steam pressure	400 lb. per sq. in.
Material of body	cast steel
Material of valve bush	bronze
Number of guide webs in valve	4.

From the proportions suggested for valves the area of the four webs works out to be 5 sq. in.

$\therefore$ area through bush = area of 6 in. diameter + 5 sq. in.

$= 28{\cdot}274 + 5 = 33{\cdot}27$ sq. in.

$\therefore$ diameter of bush $= 6\frac{5}{8}$ in.

Note that this web area is just slightly more than 15 per cent. of the pipe area.

The minimum lift for the valve will be $\frac{d}{4} = \frac{6}{4} = 1\frac{1}{2}$ in. so in order to ensure full opening the lift will be made $1\frac{3}{4}$ in. The thickness of the bush will be $\frac{7}{16}$ in. and the length of the bush will be $(2 \times 1\frac{3}{4}'' - \frac{1}{8}'') = 3\frac{3}{8}$ in. The bush flange may be $\frac{9}{16}$ in. thick and $1\frac{3}{8}$ in. wide to accommodate $\frac{5}{8}$ in. diameter studs or screws.

The dimensions of the flanges at the inlet and outlet ends of the body given in table K of the British Standards are 12 in. outside diameter, $10\frac{1}{4}$ in. pitch circle diameter, thickness $1\frac{5}{8}$ in., and drilled for 12—$\frac{7}{8}$ in. diameter bolts.

Assuming that the greatest diameter of the body is about the same as the flange diameter, viz. 12 in.

$$\text{the metal thickness } t = \frac{pd}{2f} + \text{constant}$$

$$= \frac{400 \times 12}{2 \times 7500} + 0{\cdot}25 \text{ in.}$$

$$= 0{\cdot}32 + 0{\cdot}25 = 0{\cdot}57 \text{ in. say } \tfrac{5}{8} \text{ in.}$$

The thickness of the partition metal can be made the same, and this can be increased to $\frac{3}{4}$ in. on the flat portion for the valve bush. Allowing $\frac{1}{8}$ in. for the machined face on which the bush flange fits, a thickness of $\frac{9}{16}$ in. for the cylindrical extension for the bush, and inclining the sloping metal at about 60° to the horizontal then the bush and partition metal are drawn as in fig. 16. The centre line of the body may be taken midway through the horizontal portion of the partition metal.

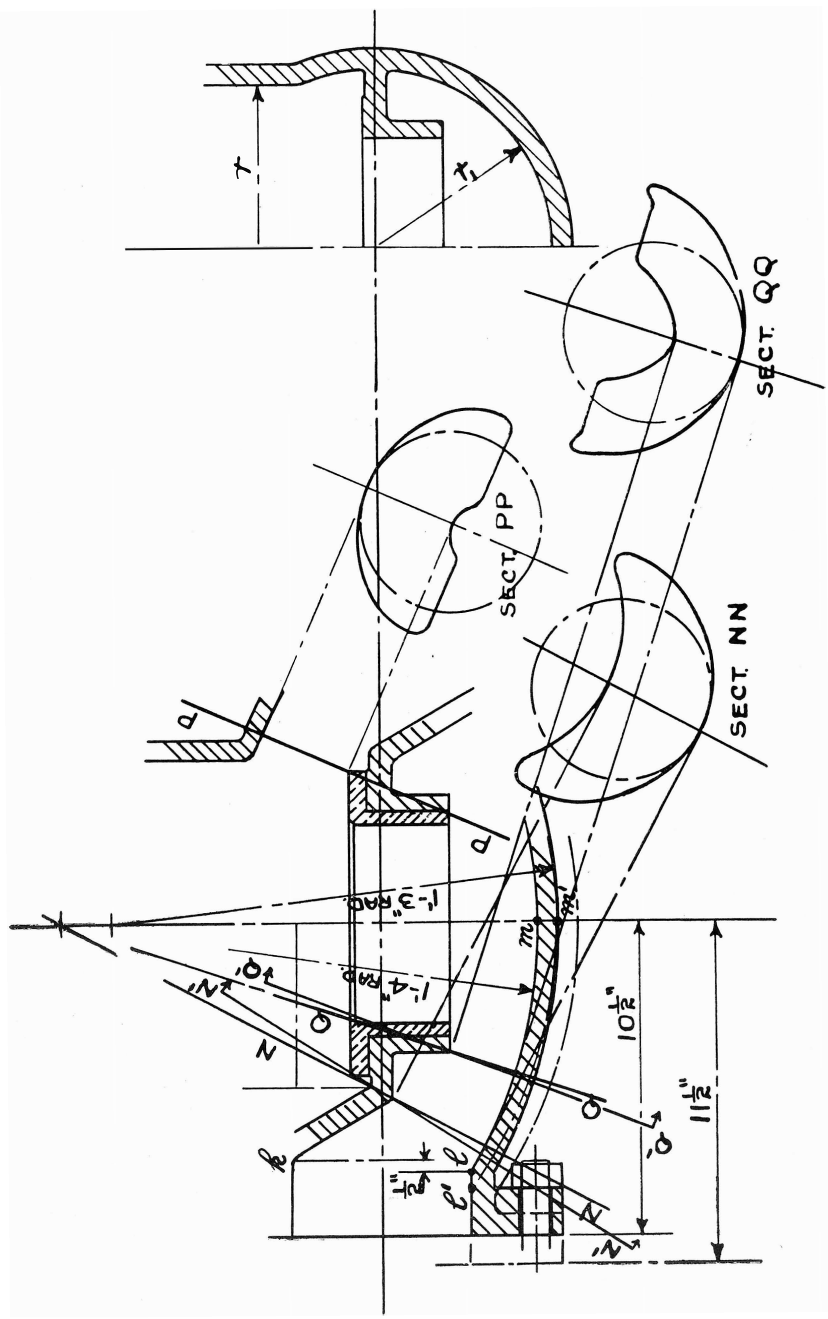


FIG. 16

Taking the largest diameter as 12 in. the inside diameter of the barrel will be $10\frac{3}{4}$ in. giving the point marked m. The sloping partition metal intersects the pipe diameter at k. To prevent contraction of area due to the filleting of the metal at k take a point l about $\frac{1}{2}$ in. to the left of k and sweep in the arc lm which will give an inside radius for the barrel about 1′–4″. To enable the nuts to be fitted on the flange bolts the outer face of the flange will require to be $10\frac{1}{2}$ in. from the centre of the valve.

Constructing the cross-sections at NN, QQ, and PP and comparing them with the circles representing the pipe area it will be seen that the areas at NN, QQ and PP are insufficient. An inspection of the drawing shows that, if we retain the same barrel diameter and raise the bush in order to increase the areas at NN and QQ, then the outlet area at PP will be decreased. The most effective method of increasing the areas at NN and QQ will be obtained by increasing the diameter of the barrel.

Increasing the inside diameter from $10\frac{3}{4}$ in. to 12 in., that is from m to m^1, and taking a point l^1 say 1 in. on the left of k, the curvature of the barrel l^1m^1 will have a radius of about 1′—3″ and the distance between the outside of the flanges will require to be increased from 21 in. to 23 in. in order to accommodate the nuts.

The sections now to be considered will be N^1N^1 and Q^1Q^1 and these can be constructed and tested for the necessary excess area.

The cylindrical portion from the barrel to the cover flange is set out in the same way as that for the right angle type of body. An inspection of the section through the centre will show that if the radius r happens to work out greater than the barrel radius r_1 a rather awkward shape would result.

Stuffing box.

The stuffing box is an integral part of the cover and its lower surface gives the starting point for outlining the cover. The proportions of the seating bush, and valve top, together with the lift of the valve, determine the distance for the bottom of the stuffing box above the facing marked F in fig. 10. When the valve is fully opened the top of the valve will be hard against the bottom of the stuffing box.

A stuffing box is shown in fig. 17. It is designed and proportioned in the following manner :—

The width of the packing space w is arbitrary and may be made equal to $0{\cdot}4\sqrt{D}$.

The depth of the packing space L will depend on the pressure in the vessel and can be made equal to—

$4w$ for pressures up to 100 lb. per sq. in.

$5w$ for pressures from 100 to 250 lb. per sq. in.

$6w$ for pressures from 250 to 500 lb. per sq. in.

$7w$ for pressures from 500 to 800 lb. per sq. in.

$8w$ for pressures from 800 to 1,200 lb. per sq. in.

The depth of the gland L^1 is made equal to $\frac{L}{2}$ so that when it is fully closed up there will be half a box of packing left to prevent leakage.

The load on the studs is calculated from the annular area of the box, plus the spindle friction, it being assumed that the full pressure acts on this area.

$$\therefore \text{ Load on studs} = \frac{p\pi\,(D_1^2 - D^2)}{4} + \text{spindle friction.}$$

The stress on the studs should not exceed 6,000 lb. per sq. in. when spindle friction is neglected, thus

$$\text{Area at bottom of threads of studs} = \frac{p \times \frac{\pi}{4}(D_1^2 - D^2)}{6{,}000 \times \text{number of studs.}}$$

For spindle diameters up to $2\frac{1}{2}$ in. two studs are usual, for diameters from $2\frac{1}{2}$ in. to 5 in. two or three studs may be used and for diameters above 5 in. four studs.

With more than two studs the gland flanges are made circular.

The thickness t of the gland flange may be made equal to the stud diameter or stud diameter plus $\frac{1}{8}$ in.

The minimum distance for the stud centres l will require to be made equal to $D_1 + 2d$ otherwise the metal of the cover might break away when drilling and tapping the stud holes.

The gland can be shaped by taking a width m of from $\frac{1}{8}$ in. to $\frac{1}{4}$ in. which will fix the radius R, then making the radius $r = d + \frac{1}{8}$ in., tangents are drawn to the two arcs.

The facing on the cover adjacent to the gland is made the same shape as the gland but $\frac{1}{8}$ in. larger all round.

The dimensions of the gland flange can be checked by considering it as a cantilever of length x. The bending moment is given by the load on one stud multiplied by the distance x.

thus external moment = internal moment of resistance

$$B.M. = fZ$$

$$\text{stud load} \times x = f\,\frac{bt^2}{6}$$

$$\text{or } f = \frac{\text{stud load} \times x \times 6}{b \times t^2}$$

The stress f for gun-metal from Data Sheet No. 2 would be 4,300 lb. per sq. in.

For example a valve for a pressure of 400 lb. per sq. in. and a spindle $2\frac{1}{2}$ in. diameter would require a packing space $w = 0{\cdot}4\sqrt{2{\cdot}5} = \frac{5}{8}$ in.

$$\text{Load on studs} = 400\,(\text{area } 3\tfrac{3}{4} \text{ in.} - \text{area } 2\tfrac{1}{2} \text{ in.})$$

$$= 400\,(11{\cdot}045 - 4{\cdot}908) = \text{say } 2{,}450 \text{ lb.}$$

$$\text{with 2 studs, load on each} = 1{,}225 \text{ lb.}$$

$$\text{area at bottom of threads} = \frac{1{,}225}{6{,}000} = 0{\cdot}204 \text{ sq. in.}$$

$$\therefore \text{ diameter of stud} = \tfrac{5}{8} \text{ in.}$$

With bolt centres $= D_1 + 2d = 3\frac{3}{4} + 1\frac{1}{4} = 5$ in. the distance x will be $\frac{5}{8}$ in. Making the radii R and r, $2\frac{1}{8}$ in. and $\frac{3}{4}$ in. respectively and drawing the shape of the gland it is found that the dimension b measures $2\frac{5}{8}$ in.

$$B.M. = fZ$$

$$1{,}225 \times \tfrac{5}{8} = \frac{4{,}300 \times 2\frac{5}{8} \times t^2}{6}$$

$$\therefore \text{ Thickness of gland } t = \sqrt{\frac{1{,}225 \times 5 \times 6 \times 8}{8 \times 4{,}300 \times 21}} = \text{say } \tfrac{11}{16} \text{ in.}$$

The neck bush is made of gun-metal and can be proportioned as shown in fig. 17. It will be seen that both the flange of the neck bush and the bottom end of the gland are champered, the angle being made about 20°. Due to this champering the packing will be forced against the spindle when the gland is being screwed up and not against the walls of the stuffing box.

The thickness of the stuffing box walls should be proportioned to suit the cover thickness and to satisfy casting considerations.

The glands for large valves and high pressure may be made of cast steel and have a gun-metal bush fitted as in fig. 18, the thickness of the bush being $0{\cdot}4\,w$ and the thickness of the flange $0{\cdot}6\,w$.

The cover is fitted to the body casting as shown in fig. 19. The thickness T of the cover may be calculated by considering it as a flat circular plate supported round the edge and with a pressure p acting uniformly over the entire surface.

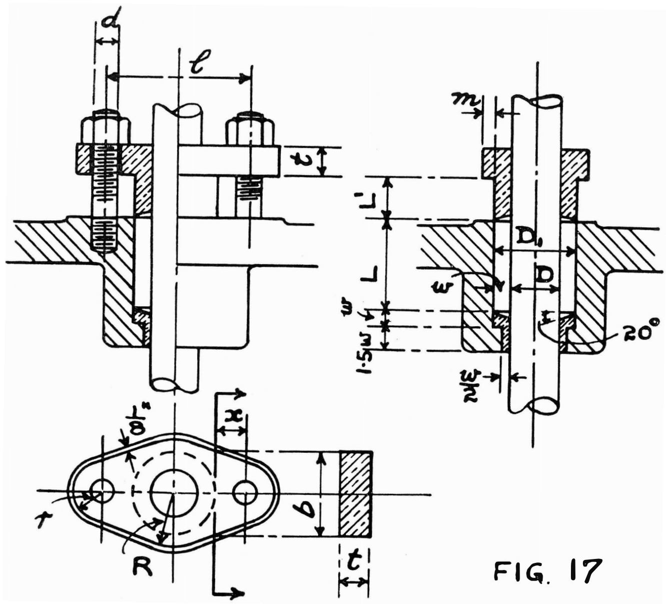


FIG. 17

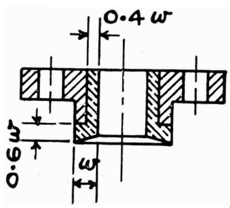


FIG. 18

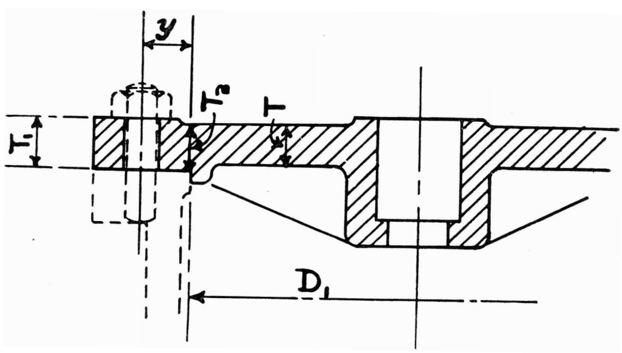


FIG. 19

The cover being rigidly bolted to the body casting more or less tends to make it fixed round the edge, but compromising on the supported and fixed condition the following formula may be taken for the design :—

$$T = D_1 \sqrt{0 \cdot 25 \frac{p}{f}} + C$$

where T = thickness in inches

D_1 = diameter of opening

p = pressure in lb. per sq. in.

f = safe working stress = 2,500 lb. per sq. in. for cast iron

= 4,300 lb. per sq. in. for gun metal

= 7,500 lb. per sq. in. for cast steel
Refer to Data Sheet No. 2.

C = casting constant = $\frac{1}{4}$ in. for cast iron

= $\frac{1}{8}$ in. for gun-metal and cast steel.

The thickness T_1 at the joint is generally made equal to the stud diameter plus $\frac{1}{8}$ in. and not less than the stud diameter.

The thickness T_2 can be $\frac{1}{8}$ in. less than T_1 and may be calculated by considering the cover subjected to a bending moment equal to the load on the cover multiplied by the distance y from the opening to the pitch circle of the studs.

The section resisting the bending is rectangular, of width equal to the circumference of the opening, and depth equal to the thickness of the cover.

$$\text{thus } B.M. = fZ = f\frac{BT_2^2}{6}$$

$$i.e. \frac{\pi D_1^2}{4} \times p \times y = f\frac{\pi D_1 T_2^2}{6}$$

$$\frac{D_1}{4} \times p \times y = f\frac{T_2^2}{6}$$

$$T_2 = \sqrt{\frac{6D_1py}{4f}}$$

Considering the same example as that taken for the practical calculations of number and diameter of studs required for bolting the cover down, the opening D_1 was $9\frac{1}{2}$ in. and the pressure p was 400 lb. per sq. in.

For the cover to be made of cast steel we have—

$$\text{Thickness } T = D_1 \sqrt{\cdot 25 \frac{p}{f}} + \tfrac{1}{8} \text{ in.}$$

$$= 9\tfrac{1}{2} \sqrt{\frac{400}{4 \times 7{,}500}} + \tfrac{1}{8}$$

$$= 1{\cdot}1 + \tfrac{1}{8} = 1\tfrac{1}{4} \text{ in.}$$

$$\text{Thickness } T_1 = \text{stud diameter} + \tfrac{1}{8} \text{ in.}$$

$$= 1\tfrac{1}{4} + \tfrac{1}{8} = 1\tfrac{3}{8} \text{ in.}$$

$$\text{Thickness } T_2 = \sqrt{\frac{6D_1py}{4f}} \qquad y = 2 \text{ in. since pitch circle of studs is } 13\tfrac{1}{2} \text{ in.}$$

$$= \sqrt{\frac{6 \times 9{\cdot}5 \times 400 \times 2}{4 \times 7{,}500}} = \sqrt{1{\cdot}52} = 1\tfrac{1}{4} \text{ in.}$$

These calculations show that the cover would require to be $1\frac{1}{4}$ in. thick, therefore to provide a machined facing for the nuts the thickness T_1 would require to be increased to $1\frac{5}{16}$ in. or $1\frac{3}{8}$ in.

This analysis takes account only of the load due to the steam pressure. Should the steam be above the valve when the valve is closed, the spindle thrust adds another factor to the conditions and complicates the problem. Any modification such as bosses required for the bridge columns, or the additional strength obtained from bridges that are cast with the cover, should amply allow for the spindle thrust.

The flange on the body casting could be made the same thickness as T_1. The holes for the studs will be drilled $\frac{1}{16}$ in. larger than the stud diameter.

A spigot should be provided on the cover where it fits into the opening so that the spindle will always remain central with the valve seating bush. The depth of the spigot is arbitrary and may vary from $\frac{1}{8}$ in. to $\frac{3}{8}$ in. The machined portion of diameter D_1 may be made one and a-half times the depth of the spigot.

Should there be a considerable difference between the outside wall of the stuffing box and the opening D_1 a number of strengthening webs can be added. These may run from the spigot to the bottom of the stuffing box as shown in fig. 19. The thickness of these strengthening webs can be made about $\frac{3}{4}\,T$.

Any modification to the cover for the attachment of the bridge or bridge columns should be made such that it will strengthen rather than weaken the cover.

Bridge.

The bridge forms the nut in which the spindle operates and may be of the following types.

(*a*) A simple forging or casting supported on two or more mild steel columns, the columns being screwed into the cover.

(*b*) Cast with the cover.

(*c*) A separate casting having flanges on the bottom for studding it to the cover.

Types (*a*) and (*b*) are the usual, type (*c*) being more suitable for valves of large diameter.

Design of type (*a*).

A sectional elevation together with a plan and a sectional end view are shown in fig. 20. The bridge and columns are of mild steel and a bronze screwed bush is fitted into the bridge. The bottom of the screwed bush should be fixed at a height h^1 above the gland studs, such that when the spindle is removed the gland can be withdrawn over the studs, *i.e.* the distance h^1 should be slightly greater than the overall depth h of the gland.

When the valve is screwed down tightly on to its seat, the spindle thrust P acts on the bridge through the flange of the screwed bush, thus the bridge acts as a beam and the columns take a tensional load.

The diameter d of the column is calculated as for a bolt and for safe design it is usual to assume that each column takes $\frac{2}{3}$ of the load. Limiting the stress at the bottom of the threads to 6,000 lb. per sq. in.

$$\text{Area at bottom of threads} = \frac{\frac{2}{3}P}{6{,}000}\,\text{sq. in.}$$

The bridge is securely tightened to the columns before the spindle is inserted, hence the shoulder formed by the difference in diameters of d and d_1 will be subjected to a crushing stress. Any elongation of the portion of diameter d that occurs when the spindle thrust P comes on the bridge will relieve this pressure, but the annular area of the shoulder has to be proportioned to take the load imposed by screwing up the nut.

$$\text{Area of shoulder} = \frac{\text{tension imposed by screwing up nut}}{\text{safe crushing stress}}$$

$$i.e.\ \frac{\pi}{4}(d_1^2 - d^2) = \frac{\text{tension imposed by screwing up nut}}{16{,}000}$$

To determine the load imposed due to screwing up a nut we have for one revolution of spanner and nut

Work done on screw = work done on spanner

$$T \times p = F \times 2\pi L \times \text{efficiency} \times S$$

$$or \quad T = \frac{F \times 2\pi L \times \text{efficiency} \times S}{p}$$

where T = equivalent tension on the screw in lb. taking into account the shear due to the twisting moment applied.
p = pitch of threads in screw in inches.
F = force applied at end of spanner in lb.
L = length of spanner in inches.
S = factor for shear.

The force F may be taken as 40 lb. or 50 lb., and the length of the spanner about 16 times the diameter of the screw. For a Whitworth vee thread an efficiency of 0·10 may be assumed. The shear factor S depends on the diameter of the screw and the following values may be taken.

Diameter of screw.	shear factor S.
$\frac{1}{2}$ in.	1·25
$\frac{5}{8}$ in.	1·25
$\frac{3}{4}$ in.	1·20
$\frac{7}{8}$ in.	1·20
1 in.	1·20
$1\frac{1}{4}$ in.	1·15
$1\frac{1}{2}$ in. and over	1·10

The columns shown in fig. 20 can be manufactured from stock bar of diameter d_1 by turning both ends down to diameter d and screwing. For a high class job a column such as that shown in fig. 21 can be used. These are machined all over and may be cylindrical of diameter d or tapered for appearance.

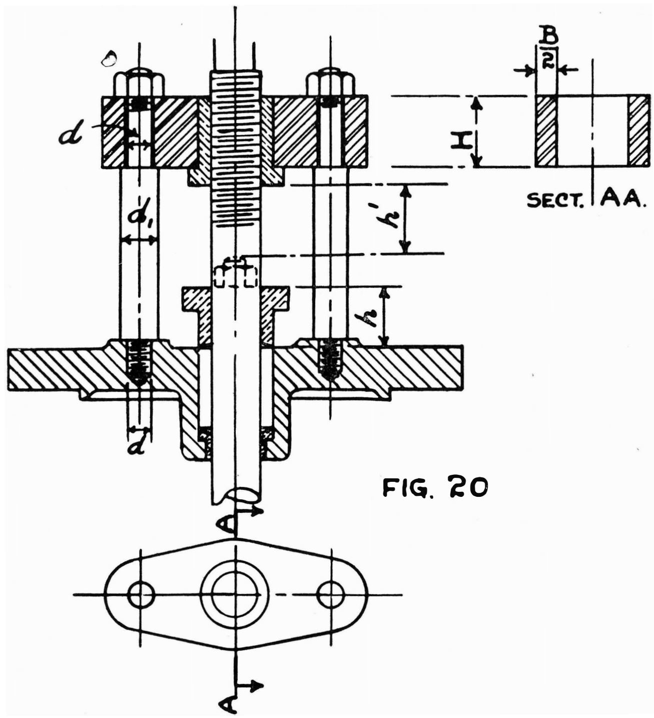


FIG. 20

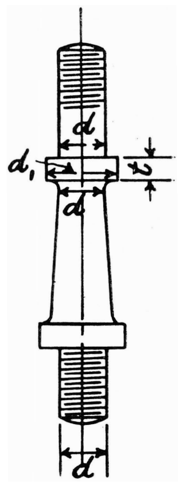


FIG. 21

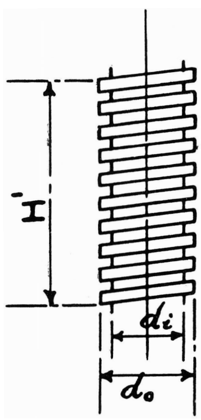


FIG. 22

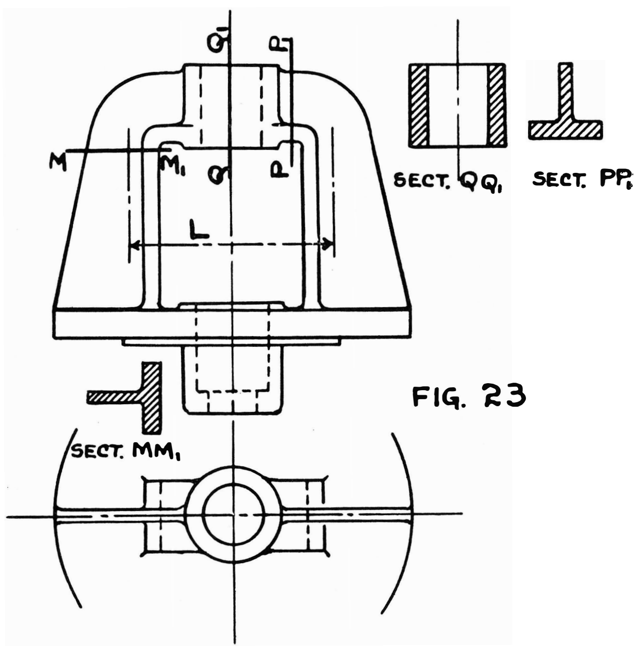


FIG. 23

When the central portion is reduced to the diameter d the thickness t of the collar forming the shoulder will require to be determined from the condition of shear.

thus

$$\text{area resisting shear} = \frac{\text{tension imposed by screwing up the nut}}{\text{safe allowable shearing stress}}$$

$$i.e. \quad \pi d t = \frac{T}{f_s}$$

$$t = \frac{T}{\pi d f_s}$$

f_s may be taken as 8,000 lb. per sq. in. for mild steel.

The columns can be screwed into the cover with a stud driver or the bottom collar can be made hexagonal to suit a standard spanner. If circular, two flats can be provided for a spanner.

The thickness of the cover may require to be modified to suit the diameter of the column screw. This can be done by putting a raised circular boss like that shown on fig. 20 either on the outside of the cover or underneath, whichever is considered best for the general appearance and manufacture. The columns should be screwed into blind tapped holes and not into the steam space.

The bridge is designed as a beam supported at the ends and loaded with a partially distributed load, the load acting over from about the middle third to the middle half of the beam. For this condition a bending moment of $\frac{PL}{5}$ may be taken. Where P = spindle thrust and L = distance between column centres. The maximum bending moment occurs at the section AA and this section is equivalent to a rectangle of width B and depth H

$$\text{hence } B.M. = fZ = f\,\frac{BH^2}{6}$$

$$i.e. \quad \frac{PL}{5} = f\,\frac{BH^2}{6}$$

values of f from Data Sheet No. 2 are—10,000 lb. per sq. in. for mild steel

7,500 lb. per sq. in for cast steel

2,500 lb. per sq. in. for cast iron.

Selecting a thickness for $\frac{B}{2}$ the depth H can be calculated. The depth H may be about four times the thickness $\frac{B}{2}$ but it must be sufficient to accommodate the screwed bush that fits into it. The length of the screwed bush is determined by the number of threads required to support the spindle thrust. The spindle screw is treated as a power screw and the length of thread obtained by taking an allowable bearing pressure of 1,000 lb. per sq. in. For a screw as in fig. 22.

Let d_o = external diameter of thread in inches

d_i = internal diameter of thread in inches

n = number of threads per inch

H^1 = screwed length of bush in inches.

P = spindle thrust in lb.

p = allowable bearing pressure per sq. in. of projected area.

$$\text{Bearing area per inch length of bush} = \frac{\pi}{4}(d_o^2 - d_i^2) \times n$$

$$\text{Bearing area for length } H^1 = \frac{\pi}{4}(d_o^2 - d_i^2) \times n \times H^1$$

$$\text{Bearing area required} = \frac{P}{p}$$

$$\therefore \frac{\pi}{4}(d_o^2 - d_i^2) \times n \times H^1 = \frac{P}{p}$$

$$\text{and } H^1 = \frac{P}{1{,}000 \times \frac{\pi}{4}(d_o^2 - d_i^2) \times n}$$

H^1 = depth of bridge metal + thickness of bush flange.

Design of type (b).

An elevation and plan together with cross-sections at MM_1 PP_1 and QQ_1 of this type are given in fig. 23. As the bridge, columns and cover form a single casting the bridge portion is considered as a fixed beam. The bending moment and the deflection diagrams for a fixed beam with a concentrated load at the centre are indicated in fig. 24. It will be seen that the stress at Q_1 is tensile, at Q compressive, near M compressive and at M_1 tensile, indicating that a tee shaped cross-section at MM_1 and PP_1 is good design theoretically as well as for the practical placing of the cover studs.

The actual bending moment is rather indefinite and for the design of the section QQ_1 a bending moment of $\frac{PL}{10}$ may be taken.

The section at MM_1 may be designed for the same bending moment, but as MM_1 is on the column portion of the casting it has in addition to the bending stress, a direct tensional stress.

The section at PP_1 is subjected to bending only and may be made the same as that designed for MM_1.

For the section MM_1 we have

$$B.M. = f_t \frac{I}{\gamma_1}$$

where P = spindle thrust

L = distance between the columns measured at the neutral axis.

$$\frac{PL}{10} = f_t \frac{I}{\gamma_1}$$

$$f_t = \frac{PL\gamma_1}{10I}$$

γ_1 = distance from the extreme fibre to neutral axis on tension side

γ_2 = distance from extreme fibre to neutral axis on compression side

and $f_c = \frac{PL\gamma_2}{10I}$

I = moment of inertia of the cross section.

f_t = stress due to bending on tension side

f_c = stress due to bending on compression side

The direct stress

$$f_d = \frac{P}{2A}$$

f_d = direct stress (tensional)

A = area of the cross-section.

The stress at $M = f_c - f_d$

The stress at $M_1 = f_t + f_d$

The problem of a suitable section may be approached as follows :—

Assuming the following data—

Spindle thrust	16,000 lb.
Spindle diameter	$2\frac{1}{2}$ in. external with 4 threads per inch (square thread)
Distance between columns ..	6 in.
Material	cast steel

The depth of the section QQ_1 to suit the screwed bush.

$$H^1 = \frac{P}{1{,}000 \times \frac{\pi}{4}(d_o^2 - d_i^2) \times n}$$

for $d = 2\frac{1}{2}$ in. and four sq. threads per inch $d_i = 2\frac{1}{4}$ in.

$$\therefore H^1 = \frac{16{,}000}{1{,}000 \times \frac{\pi}{4}(2\frac{1}{2}{}^2 - 2\frac{1}{4}{}^2) \times 4} = 4\tfrac{5}{16} \text{ in.}$$

If the bush is made $\frac{3}{8}$ in. thick the external diameter will be $2\frac{1}{2}$ in. plus $2 \times \frac{3}{8}$ in. $= 3\frac{1}{4}$ in.

The thickness of the bush flange to satisfy shear is obtained by taking a value $f_s = 3{,}000$ lb. per sq. in. for gun-metal.

$$\text{Area} = \frac{P}{f_s}$$

i.e. circumference of $3\frac{1}{4}$ in diameter $\times$ thickness $= \dfrac{16{,}000}{3{,}000}$

$$\text{and thickness} = \frac{16{,}000}{3{,}000 \times 10{\cdot}21} = 0{\cdot}52 = \tfrac{9}{16} \text{in.}$$

Depth of bridge metal $= 4\frac{5}{16}$ in. $- \frac{9}{16}$ in. $= 3\frac{3}{4}$ in.

Allowing $\frac{1}{8}$ in. at the top and on the bottom for machine facings the depth of the tee section for PP will be $3\frac{3}{4} - 2 \times \frac{1}{8} = 3\frac{1}{2}$ in.

For the section at MM_1 take the same depth, viz. $3\frac{1}{2}$ in., and assume the cross-section shown in fig. 25.

For the position of the neutral axis N.A. and the moment of inertia I. of this cross-section we have from the method of calculation given on Data Sheets Nos. 4 and 5.

sq. in.	A	n	An	I_{cc}	m	m^2	Am^2
1·875	$3 \times \frac{5}{8}$	$\frac{5}{16}$	0·586	0·061	0·7575	0·57	1·068
1·437	$2\frac{7}{8} \times \frac{1}{2}$	$2\frac{1}{16}$	2·965	0·99	0·9925	0·98	1·408
3·312			3·551	1·051			2·476
ΣA			ΣAn	ΣI_{cc}			ΣAm^2

$$\gamma_1 = \frac{\Sigma An}{\Sigma A} = \frac{3{\cdot}551}{3{\cdot}312} = 1{\cdot}07 \text{ in.}$$

$$\gamma_2 = 3{\cdot}5 - 1{\cdot}07 = 2{\cdot}43 \text{ in.}$$

$$I_{NA} = \Sigma I_{cc} + \Sigma Am^2 = 1{\cdot}051 + 2{\cdot}476 = 3{\cdot}527 \text{ in}^4.$$

Bending stress $f_t = \dfrac{P\,L\,\gamma_1}{10I}$

the length $L = 6 + 2\gamma_1$

$= 6 + 2\times 1{\cdot}07 = 8{\cdot}14$ in.

$$\therefore f_t = \frac{16{,}000 \times 8{\cdot}14 \times 1{\cdot}07}{10 \times 3{\cdot}527} = 3{,}950 \text{ lb. per sq. in.}$$

Direct stress $f_d = \dfrac{16{,}000}{2 \times 3{\cdot}312} = 2{,}420$ lb. per sq .in.

Total tensional stress on fibres at $M_1 = 3{,}950 + 2{,}420$

$= 6{,}370$ lb. per sq. in.

Bending stress $f_c = \dfrac{P\,L\,\gamma_2}{10\,I}$

$$= \frac{16{,}000 \times 8{\cdot}14 \times 2{\cdot}43}{10 \times 3{\cdot}527} = 9{,}000 \text{ lb. per sq. in.}$$

Direct stress $f_d = 2{,}420$ lb. per sq. in.

Total compressional stress on fibres at $M = 9{,}000 - 2{,}420 = 6{,}580$ lb. per sq. in.

From Data Sheet No. 2 the safe stress for cast steel would be 7,500 lb. per sq. in., thus the calculation shows that the assumed cross-section will be satisfactory for MM_1. A smaller area would satisfy for PP_1 but making the cross section at PP_1 the same as at MM_1 will allow for curvature stresses due to the small radius of curvature.

For appearance and to give a sense of stability the depth of the web is increased from MM_1 to where it joins the cover.

For the cross-section QQ_1

$$\text{B.M.} = \frac{PL}{10} = \frac{16{,}000 \times 8{\cdot}14}{10} \text{ in. lb.}$$

$$\text{B.M.} = fZ = f\frac{bd^2}{6} \text{ where } d = 3\tfrac{3}{4} \text{ ins.}$$

putting $f = 7{,}500$ lb. per sq. in.

$$b = \frac{16{,}000 \times 8{\cdot}14 \times 6}{10 \times 7{,}500 \times 3{\cdot}75 \times 3{\cdot}75} = 0{\cdot}74 \text{ in.}$$

$$\text{and } \frac{b}{2} = \frac{0{\cdot}74}{2} = 0{\cdot}37 = \tfrac{3}{8} \text{ in.}$$

This is a bit thin as the flange of the tee section is $\frac{5}{8}$ in. thick. Increasing it to $\frac{5}{8}$ in. makes the ratio $\frac{d}{b/2} = \frac{3\frac{3}{4}}{\frac{5}{8}} = 6$

and the diameter of the machined facings $= 3\frac{1}{4} + 2 \times \frac{5}{8} = 4\frac{1}{2}$ in.

Making the bush flange $4\frac{1}{4}$ in. diameter the crushing area

$= \text{area } 4\frac{1}{4} \text{ in. dia.} - \text{area } 3\frac{1}{4} \text{ in. dia.}$

$= 14{\cdot}186 - 8{\cdot}296$

$= 5{\cdot}89$ sq. in.

$$\therefore \text{ Crushing stress} = \frac{16{,}000}{5{\cdot}89} = 2{,}700 \text{ lb. per sq. in.}$$

As this stress is well on the safe side the diameter of the bush flange can be reduced to about $3\frac{7}{8}$ in.

The bush has to be fixed rigidly in the bridge so that the spindle can rotate within it. It should be a good tight fit and will require some provision to prevent it from rotating should the spindle tend to seize. If seizure occurs the full torque applied by the handwheel will have to be taken by the bush ; also, if the bush is not sufficiently tight some form of stop to take shear and crushing is necessary. With a tight fit the coefficient of friction between the metal of the bridge and bush will have a high value, and the stop used to prevent the rotation is consequently proportioned from practical considerations.

The sketches in fig. 26 illustrate a few methods showing how the bush may be fixed in the bridge.

In (*a*) Two (or more) grub screws, each screwed half in bridge and half in bush.

(*b*) A taper pin through both bridge and bush.

(*c*) A key. (After fitting, the head can be sawn off).

(*d*) The bridge metal is recessed on the underside and the bush flange has flats which fit into the recess.

(*e*) The top of the bridge metal is raised and screws having thimble points are used.

(*f*) A snug is employed, formed by screwing into the flange. The small recess in the bridge metal is cut out with a chisel.

(*g*) The bush flange is increased in diameter and fixed to the underside of the bridge with screws.

The methods shown in (*f*) and (*g*) are only suitable for bridges which are supported on M.S. columns.

Handwheel.

The handwheel may be attached to the spindle either by keying it as at (*a*) fig. 27, or making the end of the spindle square as shown at (*b*). The key is probably the stronger method, for when the spindle end is square the section may be insufficient to withstand the torque applied.

To determine the diameter of the wheel it can be assumed that a man applies a force of 25 lb. with each hand to the wheel rim in finally closing or starting to open the valve, provided of course that the wheel is not under 7 in. diameter.

By analysis similar to that given for screwing up the bridge column nuts we have for one revolution of the wheel

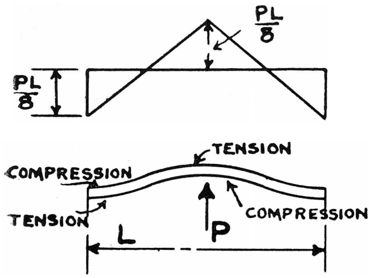


FIG. 24

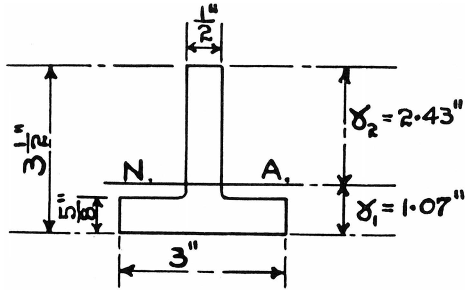


FIG. 25

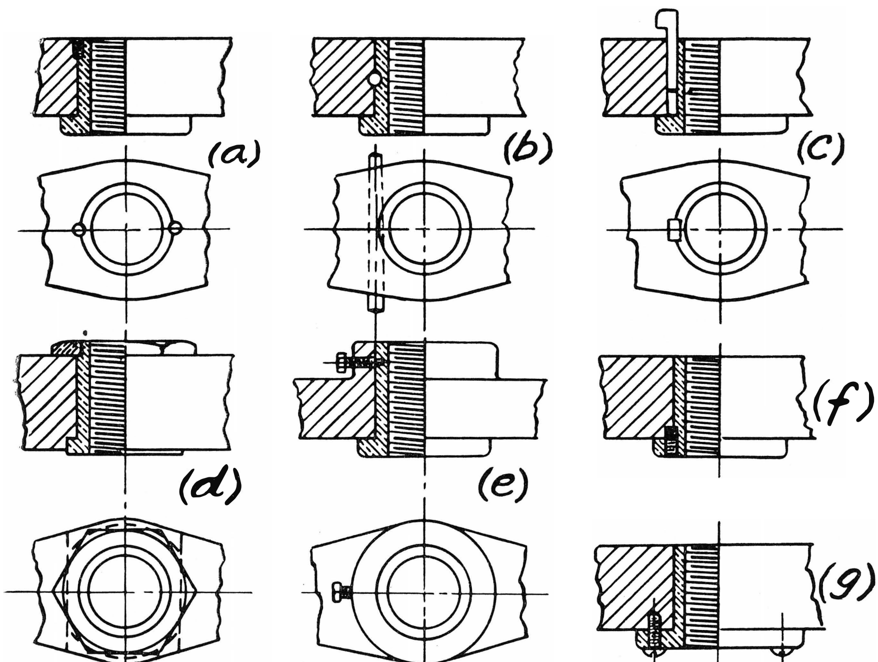


FIG. 26

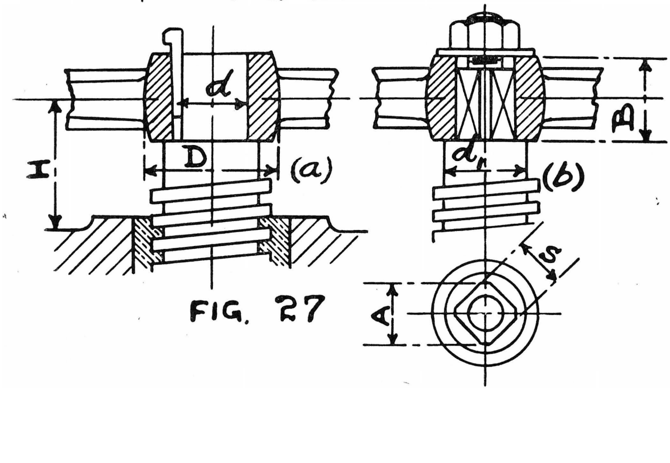


FIG. 27

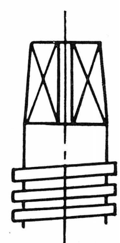


FIG. 28

Work done on spindle $=$ work done on wheel.

$$P \times p = F \times 2\pi R \times \eta$$

$$R = \frac{P \times p}{F \times 2\pi \times \eta}$$

Where $F =$ force applied at rim of wheel $= 50$ lb.

$P =$ load to close or open valve—lb.

$p =$ pitch of threads on spindle—in.

$R =$ radius of wheel—in.

$\eta =$ efficiency of screw $= 0{\cdot}10$ for a vee thread

$0{\cdot}25$ for a square thread with 4 threads per in.

$0{\cdot}20$ for 6 threads per in.

For spindles with square threads the approximate loads would be

Diameter of wheel in inches.	8	10	12	15	20
Load P in lb. with 4 t.p.i. ..	1,200	1,600	1,900	2,400	3,200
Load P in lb. with 6 t.p.i. ..	1,600	2,000	2,400	3,000	4,000

In handwheel calculations it has to be noted that on occasions a wheel spanner may be applied to the rim. This will have an effect of almost doubling the force at the rim, therefore, in calculating a suitable wheel diameter, it is not necessary to base it on the maximum spindle thrust taken for the design of the spindle, cover bolts and bridge. Satisfactory wheel diameters may be determined by taking the closing load equal to ten per cent. greater than the load due to the steam pressure.

The rims of handwheels are usually of circular cross-section and about 1 in. diameter so that they can be gripped firmly, it being difficult to grip small diameter rims. As the rim is fairly large it will transmit the load evenly to the arms, thus the arms can be designed for a $B.M. = \frac{F}{n} \times R$ where $n =$ number of arms, $R =$ radius of wheel and F = force applied at rim. Taking F = 100 lb. will allow for the use of a wheel spanner, and will give sizes for the arms that will be in general proportion to the boss and rim of the wheel.

The arms may be of elliptical or segmental cross-section and tapered from the boss to the rim, the dimensions at the rim being two thirds those at the boss.

For the size of the arms we have

$$B.M. = f\frac{I}{\gamma} = fZ$$

Z for the elliptical or segmental sections may be taken

$$\text{as } \frac{\pi}{32} tw^2 = 0{\cdot}039w^3 \text{ where } t = \text{thickness of arm} = 0{\cdot}4w$$

$$w = \text{width of arm}$$

For cast iron wheels with four arms and taking a safe working stress of 2,000 lb. per sq. in. then

$$B.M. = fZ$$

$$\frac{F}{n} \times R = f \times 0{\cdot}039w^3$$

$$\frac{100}{4} \times R = 2{,}000 \times 0{\cdot}039w^3$$

$$w = \sqrt[3]{\frac{R}{3{\cdot}12}}$$

Dimensions for arms of cast iron wheels having four arms

Diameter of wheel in.	8	10	12	15	20
Size of arm at boss	$1\frac{1}{8}'' \times \frac{9}{16}''$	$1\frac{3}{16}'' \times \frac{9}{16}''$	$1\frac{1}{4}'' \times \frac{5}{8}''$	$1\frac{3}{8}'' \times \frac{11}{16}''$	$1\frac{1}{2}'' \times \frac{3}{4}''$
Size of arm at rim	$\frac{3}{4}'' \times \frac{3}{8}''$	$\frac{13}{16}'' \times \frac{13}{32}''$	$\frac{7}{8}'' \times \frac{7}{16}''$	$\frac{15}{16}'' \times \frac{15}{32}''$	$1'' \times \frac{1}{2}$

The proportions of the wheel boss are determined from the size of the spindle end. For attachment by means of a key the diameter d is determined from the twisting moment $F \times R$.

$$\therefore\ T.M. = f_s \frac{I_o}{\gamma}$$

where I_o = polar moment of inertia of section $= \dfrac{\pi d^4}{32}$ for a circle

γ = distance from pole to extreme fibre $= \dfrac{d}{2}$ for a circle

$$\therefore\ F \times R = f_s \frac{\frac{\pi d^4}{32}}{\frac{d}{2}} = 0{\cdot}2 d^3 f_s$$

$$d = \sqrt[3]{\frac{5\,F\,R}{f_s}}$$

The key is designed for a shear load $= \dfrac{2FR}{d}$

For a spindle having a square end the square section has to satisfy the twisting moment $F \times R$.

The polar moment of inertia of a square section $= \dfrac{S^4}{6}$ where S = side of square.

The distance of the extreme fibres will be along the diagonal of the square therefore $\gamma = \sqrt{2} \times \dfrac{S}{2} = \dfrac{S}{\sqrt{2}}$

Since $T.M. = f_s \dfrac{I_o}{\gamma}$

$$F \times R = f_s \frac{\frac{S^4}{6}}{\frac{S}{\sqrt{2}}} = f_s \frac{\sqrt{2} \times S^3}{6} = f_s \frac{S^3}{4} \text{ approx.}$$

The diameter D of the boss can be made equal to $1 \cdot 8\ d$ for design (a) and $1 \cdot 8A$ for design (b).

The depth B of the boss can be made equal to $1 \cdot 25d$ for design (a) and equal to A for design (b) or, B may be made $\frac{1}{4}$ in. greater than the diameter of the wheel rim. The larger of these two dimensions should be taken.

The nut is merely to prevent the handwheel from being lifted off, and since it takes no load the size is fixed from general appearance.

The dimension A will be smaller than the diameter d_1 in order to provide a shoulder for the wheel boss. The maximum size for A would therefore be diameter d_1 minus $\frac{1}{16}$ in.

For ease in screwing the spindle up through the bridge it is advisable to make the diameter d_1 about $\frac{1}{16}$ in. smaller than the diameter at the bottom of the spindle thread. Under the heading *spindle* design it was pointed out that the handwheel torque had to be considered when fixing on a suitable diameter for the spindle. The practical attachment of the handwheel to the spindle has also an important effect in settling on a suitable diameter for the spindle.

The nut and washer in fig. 27 could be dispensed with and the squared portion tapered slightly as in fig. 28. This makes the spindle end stronger to resist the twisting moment since with the taper no shoulder is required for the handwheel boss to fit against.

The distance H from the bridge to the handwheel centre requires to be sufficient so that a man's knuckles will not be damaged against the bridge as he rotates the wheel.

Part III.

THE APPLICATION AND DESIGN OF GEARING TO OPERATE VALVE SPINDLES.

Opening and closing of large valves.

For a 6 in. diameter valve requiring a load of 16,000 lb. to close it, the radius of the handwheel would require to be

$$R = \frac{P \times p}{2\pi F \eta} \qquad \text{where } P = 16{,}000 \text{ lb.}$$

$$p = \tfrac{1}{4} \text{ in. for 4 square threads per inch}$$

$$= \frac{16{,}000 \times \frac{1}{4}}{2\pi \times 50 \times 0{\cdot}25} \qquad F = 50 \text{ lb.} \quad \eta = 0{\cdot}25$$

$$= \text{about } 50 \text{ in.}$$

A convenient diameter of wheel for applying a force of 50 lb., assuming 25 lb. with each hand would be about 10 in. or 12 in.

For a radius $R = 5{\cdot}5$ in. the force F to be applied at the wheel rim

$$F = \frac{P \times p}{2\pi R \eta} = \frac{16{,}000 \times \frac{1}{4}}{2\pi \times 5{\cdot}5 \times 0{\cdot}25} = 460 \text{ lb.}$$

thus a system of gearing is necessary, the gear ratio required being equal to $\dfrac{460}{50\eta}$ where η = efficiency of gear.

$\eta = 0{\cdot}9$ for a single spur wheel and pinion, or bevel gear

$\eta = 0{\cdot}8$ for a double gear, spur or bevel

$\eta = 0{\cdot}5$ for a worm gear.

A single spur or bevel gear would require a gear ratio of 10·2

A double spur or bevel gear would require a gear ratio of 11·5

A worm gear would require to have a reduction of 18·4.

A ratio of 10 is high for a single gear, therefore either a double gear, or a worm gear would have to be used. As 30 teeth is considered a minimum number of teeth for a worm wheel, a double threaded worm could be used.

Figs. 29 to 38 illustrate the application of gearing for operating high pressure valves of medium size, or large valves for low or medium pressure. A single spur gear is shown in figs. 29 and 30. The spur wheel is keyed to the spindle, and to allow for the vertical movement of the spindle the face of the pinion is made deeper than the wheel by an amount equal to the lift of the valve. For bridges that are cast integral with the cover, the bridge can be extended as in fig. 29 to provide for two bearings to support the pinion shaft. In order to assemble the pinion and shaft, the pinion is placed between the bearings and the shaft entered from the top, thus the top bearing requires to be larger in diameter than the bottom bearing. If d is the diameter of the shaft the diameter of the top bearing can be made $d + \frac{1}{16}$ in. and the diameter of the bottom bearing $d - \frac{1}{16}$ in. The pinion cannot be keyed to the shaft but can be pinned to it with two taper pins as shown in sketch (a) fig. 29. The pinion is extended in length and the pins fitted through the pinion and shaft on the extended portions.

Spur and pinion teeth.

The teeth of spur and pinion wheels are machine cut and of involute form. The British Standards recommend the 20° involute.

Gear wheel teeth are usually designed to satisfy both strength and wear considerations, but as valves are only operated occasionally, wear considerations can be neglected and the teeth designed for strength only. The pitch can be determined from the well known Lewis formula, and calculated on the assumption that a force of 100 lb. is applied to the handwheel rim.

If R = radius of handwheel—in.

r = radius of pinion at pitch circle—in.

P = load on pinion at pitch circle—lb.

then $P = \dfrac{100R}{r}$

From data sheet No. 7

Lewis formula :—$P = f \times p \times b \times y$

and $f = c\left(\dfrac{600}{600 + V}\right)$

The velocity V for hand operated gears is sensibly small thus f for cast iron can be taken as 8,000 lb. per sq. in.

The width b of the rim, *i.e.* the face of the wheel, is usually made from $2p$ to $3p$.

Pinion wheels with involute teeth should have not less than 16 teeth. With a small number of teeth they are undercut, *i.e.* the thickness at the root is less than the thickness at the base (involute) circle.

The following calculation shows that a cast-iron pinion about $2\frac{1}{2}$ in. diameter would be satisfactory for a design using a 12 in. diameter handwheel.

Assuming a pinion face $= 3\,p$, and a tooth factor $y =$ say $0{\cdot}104$ we have

$$P = \frac{100\,R}{r} = f \times b \times p \times y \text{ (Lewis formula)}$$

$$i.e.\ \frac{100 \times 6}{1{\cdot}25} = 8{,}000 \times 3p^2 \times 0{\cdot}104$$

$$p = \sqrt{\frac{100 \times 6}{1{\cdot}25 \times 8{,}000 \times 3 \times 0{\cdot}104}} = 0{\cdot}44 \text{ in.}$$

From data sheet No. 6 a 7 *DP* tooth has a circumferential pitch $= 0{\cdot}449$ in.

Number of teeth in pinion $= 7 \times 2\frac{1}{2} = 17\frac{1}{2}$

$\therefore$ make pinion with 18 teeth and pitch circle diameter $= \dfrac{18}{7} = 2{\cdot}571$ in.

Pinion shaft.

The pinion shaft is subjected to bending and twisting. Combining the bending moment and twisting moment into an equivalent twisting moment we have by the Guest formula

$$\text{equiv. } T.M. = \sqrt{B.M.^2 + T.M^2}.$$

$$\text{for a } B.M. = T.M. \text{ then equiv. } T.M. = \sqrt{2} \times T.M.$$

In normal designs as in figs. 29 and 30 the *T.M.* is probably the greater moment but for safety in design the shafts can be proportioned for an equiv. $T.M. = \sqrt{2} \times T.M.$

$$\therefore \sqrt{2} \times T.M. = f_s \frac{I_o}{\gamma}$$

where $f_s =$ safe shearing stress $= 5{,}000$ lb. per sq. in. (Guest formula)

$$I_o = \text{polar moment of inertia} = \frac{\pi d^4}{32}$$

$$\gamma = \text{radius of shaft} = \frac{d}{2}$$

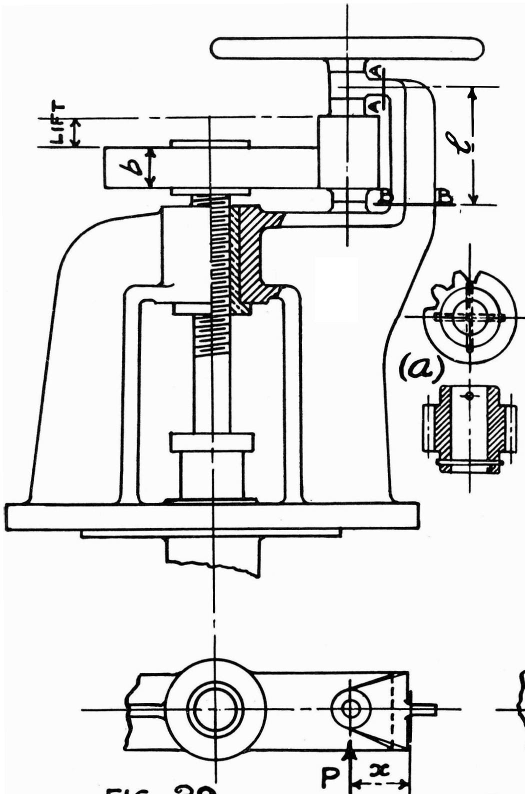


FIG. 29

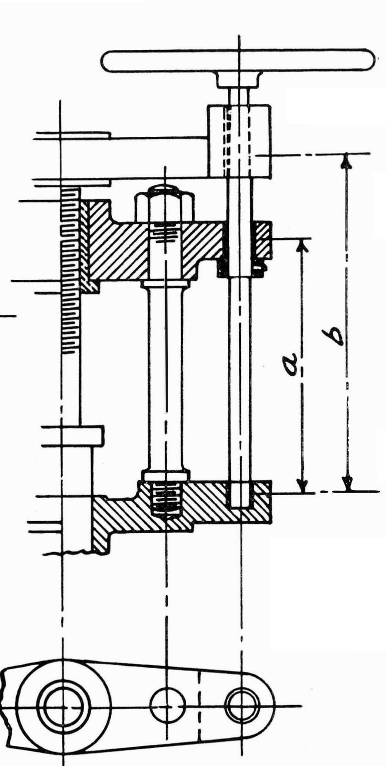


FIG. 30

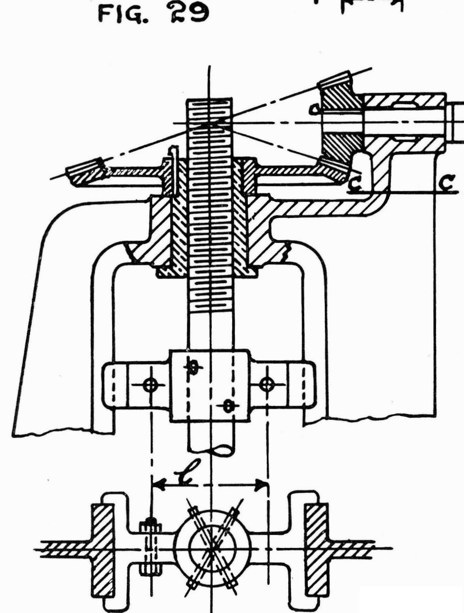


FIG. 31

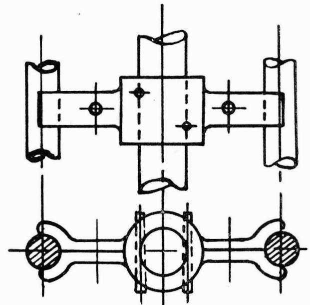


FIG. 31a

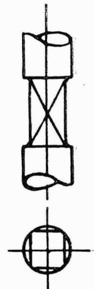


FIG. 31b

Designing for a force of 100 lb. applied to a handwheel rim of radius R

$$\text{then } \sqrt{2} \times 100 \times R = 5{,}000 \times \frac{\dfrac{\pi d^4}{32}}{\dfrac{d}{2}}$$

$$\sqrt{2} \times 100 \times R = 5{,}000 \times 0{\cdot}2d^3$$

$$d = \sqrt[3]{\frac{\sqrt{2} \times 100 \times R \times 5}{5{,}000}}$$

$$= 0{\cdot}52\sqrt[3]{R}.$$

The diameter thus determined should be increased for the keyway if the pinion is keyed to the shaft. With a pinion pinned to the shaft as at (*a*) fig. 29 the shaft will require to be larger in diameter. The diameter of the pins may be one quarter of the diameter of the shaft, if larger too much area is taken from the shaft.

Bearings.

The load on the bottom bearing will be a little greater than that on the top bearing, as the maximum load is applied when the wheels are in mesh at the lower part of the pinion face.

Since load on pinion teeth $P = \dfrac{100R}{r}$

then for a 12 in. handwheel and a 3 in. pinion

$$P = \frac{100 \times 6}{1{\cdot}5} = 400 \text{ lb.}$$

of which the lower bearing may take 250 lb. and the upper bearing 150 lb.

With an allowable bearing pressure per sq. in. of projected area $p = 200$ lb. we have

$$\text{Projected area of bearing} = \frac{\text{Load}}{p} = \frac{250}{200} = 1{\cdot}25 \text{ sq. in.}$$

$$\text{Projected area} = \text{diameter} \times \text{length} = 1{\cdot}25$$

$$\therefore \text{length} = \frac{1{\cdot}25}{\text{diameter}}$$

since $d = 0{\cdot}52\sqrt[3]{R} = 0{\cdot}52\sqrt[3]{6} = 0{\cdot}944$ in. then to allow for a keyway the pinion shaft can be made $1\frac{1}{8}$ in. diameter.

$\therefore 1\frac{3}{16}$ in. diameter at top bearing and $1\frac{1}{16}$ in. diameter at bottom bearing

$$\text{thus length} = \frac{1{\cdot}25}{1{\cdot}0625} = 1{\cdot}176 \text{ in. or } 1\tfrac{3}{16} \text{ in.}$$

The length of the upper bearing would be less but since the length of a bearing should not be less than the diameter, the upper bearing would be made say $1\frac{1}{4}$ in. long.

Extension of bridge metal to upper bearing.

The cross-section at BB is subjected to a combined stress due to bending and twisting. The bending moment is $P \times l$ and the twisting moment $P \times x$ where $P =$ load on upper bearing. In designing the section BB it will be sufficient to design for bending only and to take a $BM = 1{\cdot}25P \times l$ to cover the additional stress due to the twisting.

The cross-section at AA is subjected to bending only.

To obviate the twisting, the bridge extension could be made as shown in fig. 33 in which the upper bearing is a separate casting that can be attached to the bridge metal with four set screws. This separate casting would enable the pinion to be keyed to the shaft.

The design given in fig. 30 is suitable for bridges supported on M.S. columns, and will allow for the pinion to be keyed to its shaft.

The pinion shaft is designed for bending and twisting, and is considered as a supported beam loaded outside the supports, thus the bearing loads are—

$$\text{Upper bearing} = \text{force at } P.C. \text{ of pinion } \times \frac{b}{a}$$

$$\text{Lower bearing} = \text{force at } P.C. \text{ of pinion } \times \frac{b - a}{a}$$

and the bending moment $=$ force at $P.C.$ of pinion $\times$ $(b - a)$.

To prevent the shaft from lifting out, a collar attached to the shaft by a grub screw can be placed under the bridge. If the bridge is made of M.S. a gun-metal bush would be necessary for the upper bearing, if made of cast iron the bush could be dispensed with.

The application of a bevel gear is shown in fig. 31. The bevel wheel cannot move vertically like the spur wheel, therefore it must rotate in the bridge and thus act as a rotating nut. The spindle will require to be prevented from rotating by some suitable guide fixed to the spindle like that shown in fig. 31 for bridges cast integral with the cover, or as in fig. 31a when round M.S. columns are used. In the design shown in fig. 31, a gun-metal screwed bush rotates in the bridge casting and the wheel is keyed to the bush, hence, as the bush rotates the spindle moves up or down to open or close the valve.

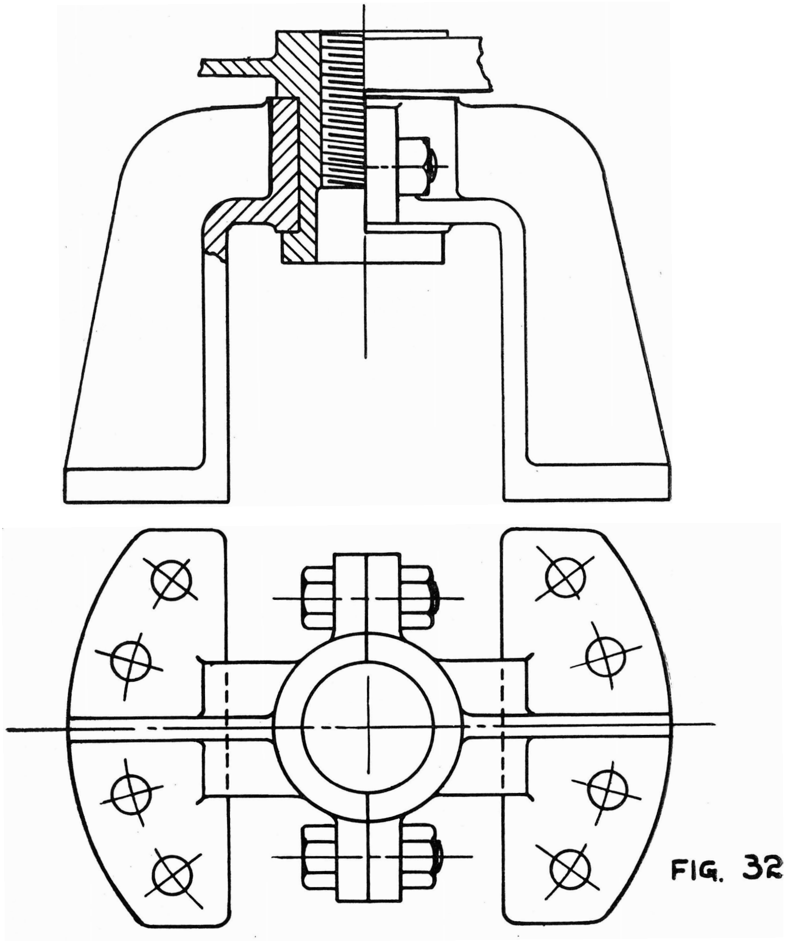


FIG. 32

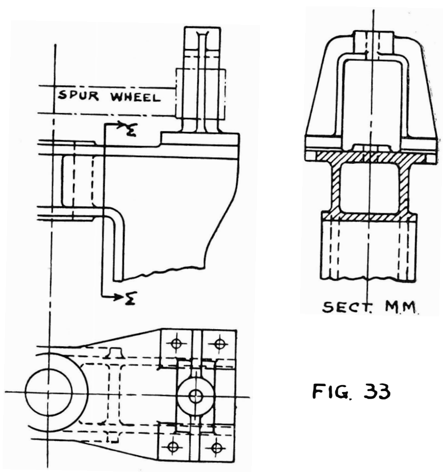


FIG. 33

An alternative design is shown in fig. 32 in which the wheel boss is lengthened and machined to suit, the wheel being screwed internally to act as the nut. For the practical assembling of the wheel the bridge requires to be cast in two parts, bolted together as shown and studded to the cover.

In fig. 31 the bearing carrying the bevel pinion shaft is made long, and two gun-metal bushes fitted thus making it equivalent to a two-bearing support. The shaft is designed for twisting and allowance made for the keyway. The end of the shaft is enlarged in diameter then squared off to take the handwheel, or a handle as shown at (h) may be fitted. The spindle guide is made in two parts, bolted together, then attached to the spindle by tapered pins. Pins fitted as in fig. 31a do not weaken the spindle as much as when they pass through the axis. Alternatively the spindle could be squared off as shown in fig. 31b, keeping the width across the flats equal to the diameter at the bottom of the spindle thread. This does not weaken the spindle since the designed diameter of the spindle is that at the bottom of the threads. This method also has the advantage that no pins are necessary. The guide may be designed to take 75 per cent. of the maximum torque applied to the spindle therefore the load on each bolt will be equal to $0{\cdot}75 \times P \times R$ divided by $\frac{l}{2}$ where $P =$ load on bevel wheel teeth and $R =$ mean radius of the bevel wheel. The section at CC is designed in a similar manner to that for section BB, fig. 29.

The application of a double spur gear is shown in fig 34, and for a combined bevel and spur gear in fig. 35. It will be observed that the design in fig. 35 is more compact than that in fig. 34 since the wheel on shaft B in the double spur arrangement has to be fairly large in diameter, in order that the handwheel shaft will clear the spur wheel on the valve spindle. An advantage of the double gear arrangement is that, when closing the valve the handwheel can be put on shaft B for rapid closing, then removed, and put on shaft A to obtain the required closing pressure. Similarly when opening, the handwheel is placed on shaft A and the valve eased off the face, then, when the pipe is full and the valve in equilibrium, the handwheel is removed to shaft B for more rapid opening. Alternately two handwheels may be fitted. For ease in assembling the shaft A in fig. 35, one bearing is made in the bridge casting, the other being separate and bolted to the bridge casting with two set screws.

The application of a worm gear is shown in fig. 36. Like that of the bevel gear the wheel acts as a nut and the spindle does not rotate. The bearings A and B have to take the thrust of the worm thus the section at dd is subject to a bending moment $= T \times x$ where $T =$ the tangential force on the gear. The bridge can be modified as shown so that the bearings can be cast with it, or the bearings may be split bearings of the plummer block type. Separate bearings would permit the worm to be made of mild steel and forged integral with the shaft. With solid bearings as shown the worm would be pinned to the shaft. The couple $T \times r$ is balanced by the couple $P \times L$ thus the bearings have to be designed for a radial load P and a thrust T.

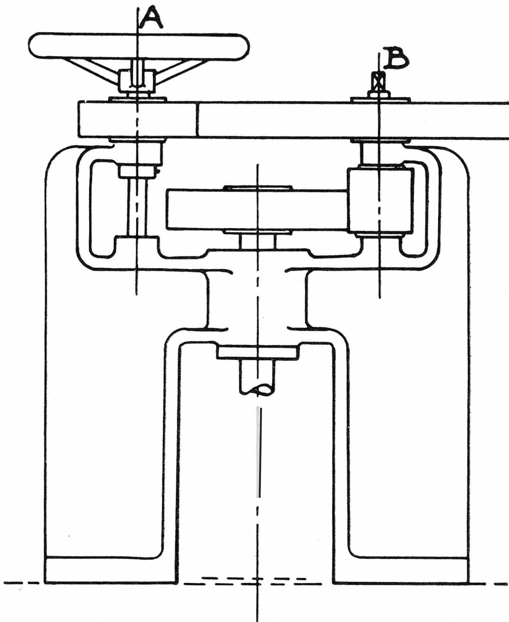


FIG. 34

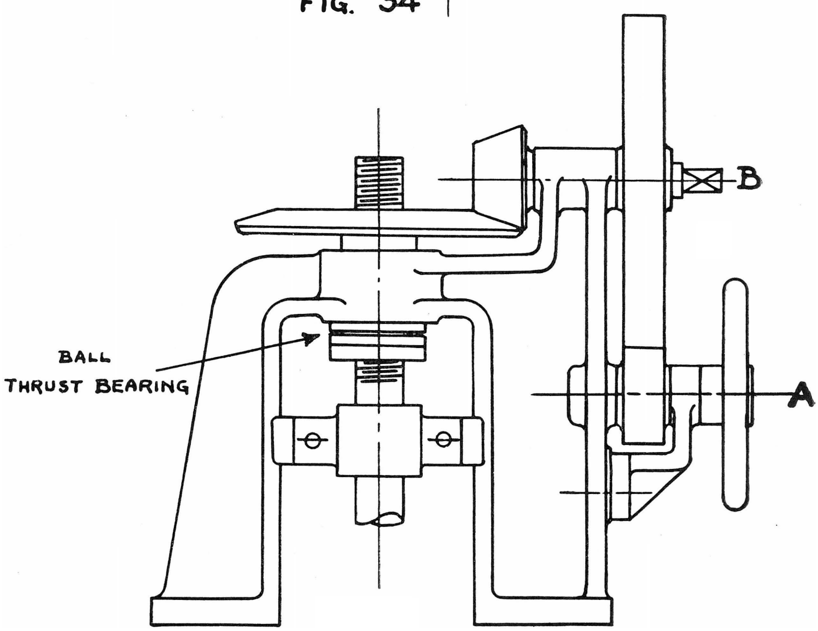


FIG. 35

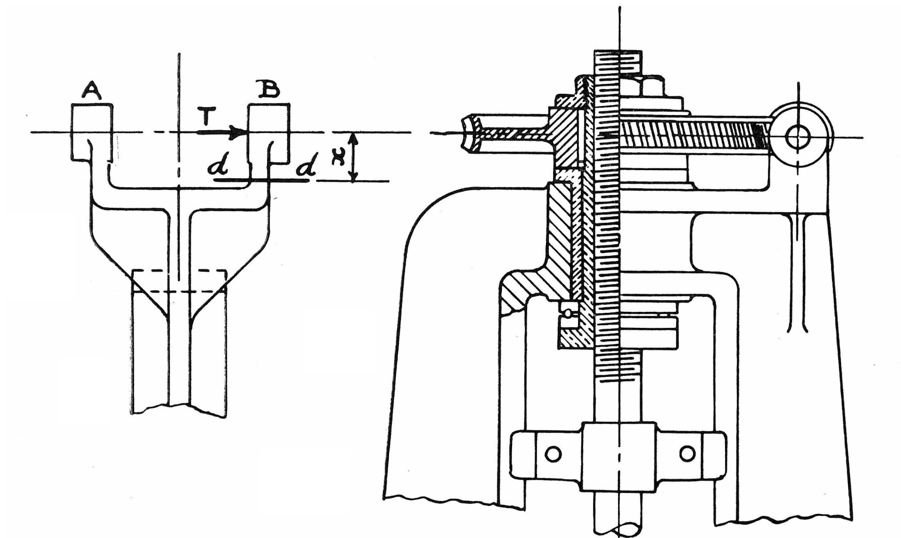


FIG. 36

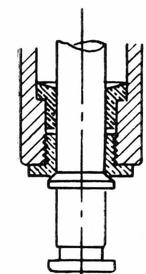
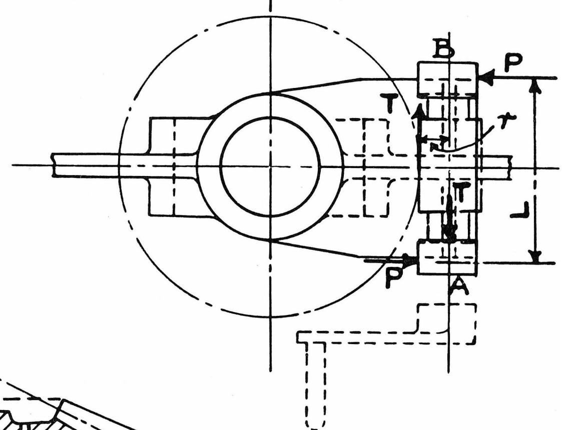


FIG. 37



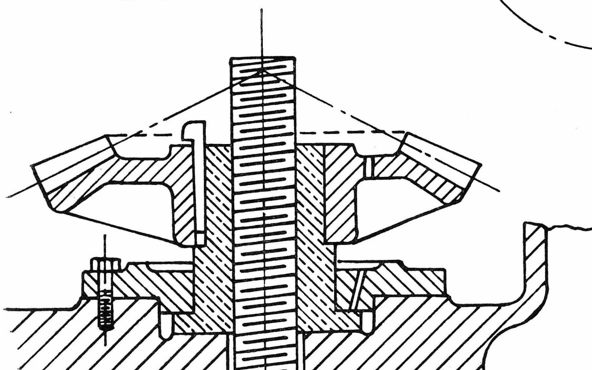


FIG. 38

Method of determining the dimensions of a worm gear.

Taking the case of a worm gear suitable for a spindle $2\frac{1}{2}$ in. external diameter which is screwed 4 threads per inch, square thread, and the load to close the valve being 16,000 lb.

Assume a handle of radius 5 in. and that for easy operation it can be operated by a force of 25 lb. applied with one hand.

In one revolution of the worm the circumference of the wheel will move through a distance equal to the lead of the worm thread thus

work done on wheel = work done on handle

$$\frac{T \times l}{\eta} = F \times 2\pi \times R$$

$$i.e.\ T \times l = F \times 2\pi \times R \times \eta$$

where T = tangential force on worm wheel in lb.

l = lead of the worm in inches

F = force applied to handle in lb.

R = radius of handle in inches

η = the efficiency of the gear.

The efficiency η may be determined from either of the following formulae—

$$\eta = \frac{\cos\delta \sin 2\alpha}{\cos\delta \sin 2\alpha + 2\mu} \quad \text{or} \quad \eta = \frac{\tan\alpha}{\tan(\alpha + \phi)}$$

where δ = the pressure angle of the involute tooth = 20° recommended by the British Standards.

α = the lead angle

μ = the coefficient of friction = 0·05 oiled condition

= 0·25 dry condition

ϕ = the angle of friction between worm and wheel

i.e. $\tan\phi$ = the coefficient of friction μ

Taking $\delta = 20°$, $\alpha = 9°$, and $\eta = 0{\cdot}15$ for average conditions

$$\text{the efficiency } \eta = \frac{\cos 20° \sin 18°}{\cos 20° \sin 18° + 0{\cdot}3}$$

$$= \frac{0{\cdot}9397 \times 0{\cdot}309}{0{\cdot}9397 \times 0{\cdot}309 + 0{\cdot}3} = 0{\cdot}492 = 49{\cdot}2\%$$

thus an efficiency of 0·5 may be taken for general design purposes.

Tangential force on wheel.

$$T = \frac{F \times 2\pi \times R \times \eta}{l}$$

$$= \frac{25 \times 2\pi \times 5 \times 0{\cdot}5}{l} = \frac{390}{l}$$

Load on wheel teeth.

This is a function of the tangential force, the pressure angle, and the lead angle and is given by the following—

$$\text{Load on teeth} = \frac{T}{\cos \delta \cos \alpha}$$

It is generally assumed for single and double threaded worms that the load is taken by two teeth as there are probably two or more teeth in mesh at any instant.

$$\therefore \text{Load on each tooth } P = \frac{T}{2 \cos \delta \cos \alpha}$$

$$\text{or } P = \frac{390}{l \times 2 \times \cos 20^\circ \cos 9^\circ}$$

$$= \frac{390}{l \times 2 \times 0{\cdot}9397 \times 0{\cdot}9877} = \frac{210}{l}$$

for a single thread worm $l = p$ where $p =$ pitch

for a multiple thread worm $l = np$ where $n =$ number of threads or starts

$$\text{hence for a single thread worm } P = \frac{210}{l} = \frac{210}{p}$$

$$\text{and for a double thread worm } P = \frac{105}{p}$$

The safe tooth load P by the Lewis formula as given for the design of spur gearing is

$$P = f \times p \times b \times y$$

For cast iron wheels at low speed f may be taken as 7000 lb. per sq. in then assuming $b = 2p$ and $y = 0{\cdot}12$

$$P = 7{,}000 \times p \times 2p \times 0{\cdot}12$$

$$= 1{,}680p^2$$

$$\therefore \text{ with a single thread worm } \frac{210}{p} = 1680\, p^2$$

$$i.e. \qquad p^3 = \frac{210}{1680} \text{ or } p = 0{\cdot}5 \text{ in.}$$

$$\text{With a double thread worm } \frac{105}{p} = 1{,}680p^2$$

$$i.e. \qquad p^3 = \frac{105}{1{,}680} \text{ or } p = 0{\cdot}397 \text{ in.}$$

Tangential load on wheel and thrust on worm shaft bearings

$$T = \frac{390}{l} = \frac{390}{p} = \frac{390}{0{\cdot}5} = 780 \text{ lb. for single thread worm}$$

$$= \frac{390}{l} = \frac{390}{2p} = \frac{390}{0{\cdot}794} = 490 \text{ lb. for double thread worm.}$$

Diameter of worm wheel.

In 1 revolution of wheel

work done on valve spindle = work done on wheel

$$Wp = T \times \pi D \times \eta$$

$$\therefore \text{ Diameter of wheel } D = \frac{W \times p}{T \times \pi \times \eta}$$

where $W = 16{,}000$ lb.
$p = \frac{1}{4}$ in. (pitch of spindle thread)
$\eta = 0{\cdot}25$ (the efficiency of spindle thread)

$$\therefore D = \frac{16{,}000 \times \frac{1}{4}}{780 \times \pi \times 0{\cdot}25} = \text{about } 6\tfrac{5}{8} \text{ in. for single thread worm}$$

$$= \frac{16{,}000 \times \frac{1}{4}}{490 \times \pi \times 0{\cdot}25} = 10 \text{ in. for double thread worm.}$$

Adopting a standard pitch for the wheel teeth from data sheet No. 6 we would require a 6 D.P. tooth for the single worm and an 8 D.P. tooth for the double worm giving

Number of teeth in worm wheel	Revolutions of handle per revolution of worm wheel	
40	40	for single worm
80	40	for double worm

If the lift of the valve is $1\frac{3}{4}$ in. then 7 complete revolutions of the worm wheel are required which means 280 turns on the handle. This would make the full opening and closing rather a slow process, but if the valve spindle thread is made double instead of single the turns on the handle would be reduced to 140, further if the size of the worm wheel teeth is increased without increasing the wheel diameter the turns of the handle will be still fewer.

Making the valve spindle thread double and taking a double thread worm and a 10 in. diameter worm wheel with teeth of 6 D.P. we have for a $1\frac{3}{4}$ in. valve opening

$$\text{rev. of handle} = \frac{1\frac{3}{4}}{\frac{1}{2}} \times 30 = 105$$

since lead of spindle thread $= \frac{1}{2}$ in. and number of teeth in worm wheel $= 6 \times 10 = 60$

The larger tooth would be an advantage as a larger handle could be used, also a greater force than 25 lb. could be applied if required, either of which means an increase of tooth load.

Since the tangential force $T = \dfrac{390}{l}$ the larger tooth and consequent lead results in a smaller thrust on the bearings for the same torque applied.

Manipulating would be fairly easy as the maximum load has only to be applied at the initial opening of the valve and at the final closing.

For a worm having more than two threads, the size of tooth, diameter of worm and wheel, and revolutions of worm required for full valve opening, may be studied in a similar manner.

Diameter of worm.

Let α = lead angle (taken as 9° in above calculation)

$$\text{then } \tan \alpha = \frac{\text{lead}}{\text{circumference of worm pitch circle}}$$

$$\textit{i.e.}\ \tan 9^\circ = \frac{l}{\pi d} \therefore d = \frac{l}{\pi \tan 9^\circ} = \frac{l}{3{\cdot}14 \times 0{\cdot}158} = 2{\cdot}016l$$

for a 6 D.P. tooth $l = 2p = 2 \times 0{\cdot}524 = 1{\cdot}048$

$$\therefore d = 2{\cdot}016 \times 1{\cdot}048 = 2{\cdot}113 = \text{say } 2\tfrac{1}{8} \text{ in.}$$

Length of worm.

$$\text{length} = 10 \times \text{module, or about } 4 \times \text{pitch}$$

$$\text{module} = \frac{1}{\text{D.P.}} = \tfrac{1}{6} \text{ in.}$$

$$10 \times \text{module} = \text{say } 1\tfrac{3}{4} \text{ in.}$$

$$\text{pitch} = 0{\cdot}524$$

$$4 \times \text{pitch} = \text{say } 2 \text{ in.}$$

$\therefore$ make length not less than $1\frac{3}{4}$ in. or greater than 2 in.

The designs described presume that a moderate effort is applied to a handle or wheel for opening and closing stop valves, but should the spindle have to make a large number of turns for full opening, and a quicker opening is desired, an electric motor can be fitted.*

Sealing stuffing boxes against pressure.

The packing in the stuffing box may be protected from the steam in high pressure valves by modifying the bottom of the spindle to provide a valve as indicated in fig. 37. A bronze bush can be fitted into the bottom of the casting to form a valve seat. The bush could be screwed into the casting or fixed to it by two or three gun-metal screws through the flange. This valve seals off the stuffing box when the valve is full open.

Designs to be practical.

It is important that the component parts of any operating gear can be made in the workshops and the whole design such, that the parts can be easily assembled and dismantled for overhaul and repair.

Lubrication.

Provision must be made for lubricating all moving parts. The sketches given in this treatise do not include the method of lubrication, but an alternative design to that given in fig. 31 is shown in fig. 38 from which it will be seen that the flange on the rotating bush can be satisfactorily lubricated.

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* For an illustration of a 7-inch diameter valve for a pressure of 1,320 lb. per sq. inch with worm gearing, the worm being operated by an electric motor—see "Engineering," February 21st, 1930.

Part IV.

THE DESIGN OF CURVED MACHINE PARTS.

Design of curved machine members.

Many machine parts are subject to compound stress such as bending combined with a direct tensile or compressive stress. Cast bridges are examples of compound stress. The treatment already given for the design of cast bridges is based on the straight beam theory but when machine members are curved they should be proportioned to satisfy stresses determined by the curved beam theory.

Fig. 39 indicates a curved frame for housing the worm of a worm gear mechanism together with diagrams showing the maximum fibre stresses for the section MN as determined by the straight and curved beam analyses.

Straight beam analysis.

Let C be the centroid of the cross-section MN distant γ_1 from M, and γ_2 from N, and let T be the thrust of the worm. At C introduce two equal and opposite forces equal to T. The force T at the worm axis together with the force T at C acting towards the fixing flange of the casting form a couple which tends to rotate the casting in the direction shown. The cross-section at MN is therefore subject to a bending moment $= T \times l$ hence the fibres at M are in tension and the fibres at N in compression. The force T at C acting away from the fixing flange puts a direct tension over the whole cross-section at MN.

Maximum stress at M.

From 2nd fundamental formula $B.M. = f\dfrac{I}{\gamma}$

$$\text{the maximum bending stress at } M = \frac{B.M. \times \gamma_1}{I}$$

From 1st fundamental formula $T = Af_t$

$$\text{the direct stress} = \frac{T}{A}$$

$\therefore$ Total stress on fibres at M = bending stress + direct stress.

$$= \frac{B.M. \times \gamma_1}{I} + \frac{T}{A}$$

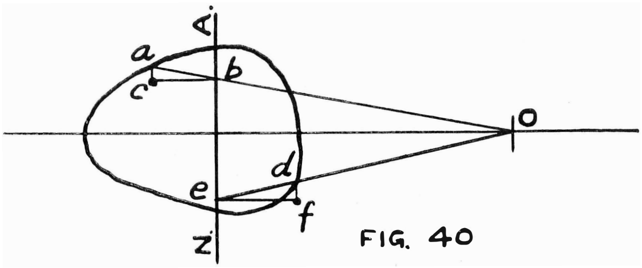

FIG. 40

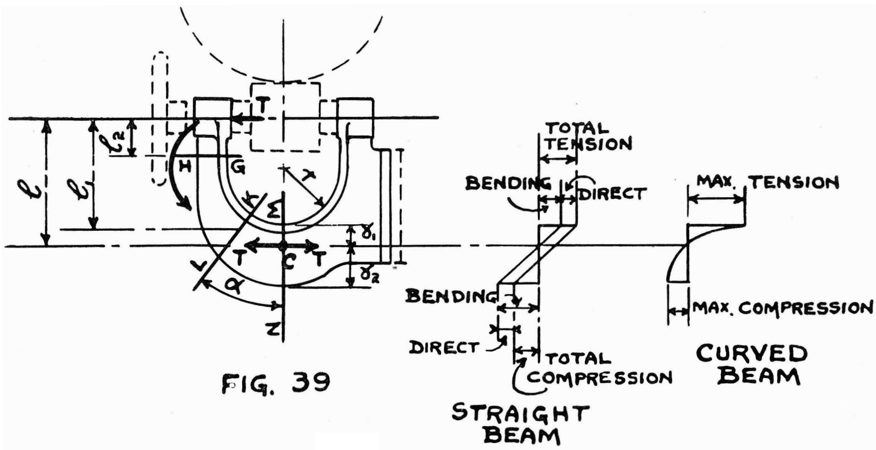

FIG. 39

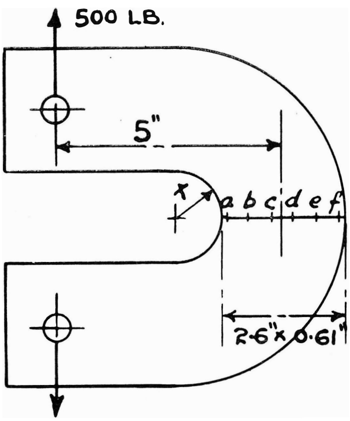

FIG. 41

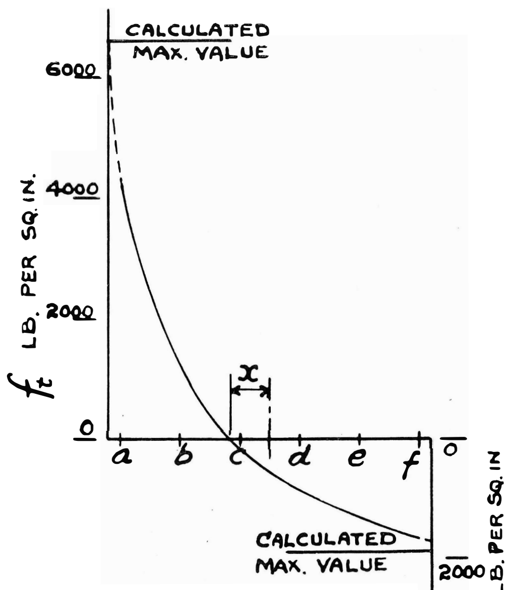

FIG. 42

Maximum stress at N.

The bending stress at N is compressive, and the direct stress tensile.

$$\therefore \text{ Total stress on fibres at } N = \frac{B.M. \times \gamma_2}{I} - \frac{T}{A}$$

Curved beam analysis.

The extreme fibre stresses depend on the radius of curvature and may be calculated by the following formulæ*

$$\text{Maximum stress at } M = \frac{T}{A}\left[1 + \frac{l}{R - \gamma_1}\left(\frac{\gamma_1}{R} \times \frac{A^1}{A^1 - A} - 1\right)\right]$$

Where T = thrust or load in lb.

l = distance of load from the neutral axis of the cross-section in inches

A = area of cross-section in sq. in.

γ_1 = distance of extreme tension fibre to neutral axis

γ_2 = distance of extreme compression fibre to neutral axis.

R = radius of curvature to neutral axis.

A^1 = modified area in sq. in.

$$\text{Maximum stress at } N = \frac{T}{A}\left[1 + \frac{l}{R + \gamma_2}\left(- \frac{\gamma_2}{R} \times \frac{A^1}{A^1 - A} - 1\right)\right]$$

The modified area A^1 can be determined graphically as follows for any cross-sectional area.

In fig. 40 let O be the centre of curvature and $N.A.$ the neutral axis. Take any point a on the perimeter of the cross-section and draw aO cutting $N.A.$ at b. From a draw a line parallel to $N.A.$ and from b a line perpendicular to $N.A.$ These lines intersect at c and give a point on the perimeter of the figure the area of which gives the modified area. Taking a number of points the whole figure is obtained, *e.g.* from point d on the cross-section the point f is obtained.

* Refer to Machine Design Drawing Room Problems by Albert.

Another method adopted by designers for calculating the maximum fibre stresses on curved frames is to calculate the bending stress as for a straight beam and to multiply this stress by a coefficient k to allow for the curvature.

Thus

Maximum stress on tension fibres, *i.e.* at M = direct stress + bending stress $\times\ k$

$$= \frac{T}{A} + \frac{B.M. \times \gamma_1}{I} \times k$$

The following formula for k is suggested by Wilson and Quereau in University of Illinois circular, No. 16, 1928.

$$k = 1{\cdot}00 + 0{\cdot}5 \frac{I}{b\gamma^2}\left[\frac{1}{R-\gamma} + \frac{1}{R}\right]$$

Where I = moment of inertia of cross-section

b = maximum width of cross-section

γ = distance from $N.A.$ to extreme fibre on concave side

R = radius of curvature to $N.A.$

For circular and elliptical cross-sections use 1·05 instead of 0·5.

The designer is concerned with maximum stresses and these can be determined by either of these formulæ.

For the cross-section at KL on the curved part the $BM = T \times l_1$ and the direct stress $= \dfrac{T \cos a}{\text{area at } KL}$

The cross-section GH on the straight part has no direct stress and is designed for a $B.M. = T \times l_2$ by the straight beam analysis.

Experimental results for a curved frame.

In order to find out how the stress varied between the two extreme fibres of a curved frame, the author made a few experiments in the laboratory at the University of Queensland with simple cast iron frames curved with different radii. All the specimens were cast with the same mixture. Bars were also cast with the same mixture in order to determine an average value for Young's modulus for the material. The bars were tested as beams, supported at the ends on knife edge supports. Loads were placed at the centre and the deflections were carefully measured.

For this condition

$$\delta = \frac{WL^3}{48EI} \text{ hence } E = \frac{WL^3}{48 \times I \times \delta}$$

where W = load in lb.

L = length between supports in inches

I = moment of inertia of cross-section—in.^{4}

E = Young's modulus—lb. per sq. in.

δ = deflection in inches.

From the results recorded

An average figure of $13 \cdot 7 \times 10^6$ lb. per sq. in. for E was obtained.

The frames, each with a different radius r were of the shape and dimensions shown on fig. 41. A load of 500 lb. at a distance of 5 in. from the $N.A.$ was applied. The strains due to this load were measured by attaching Huggenberger tensometers to the specimens at the points a, b, c, d, e, and f.

Since $E = \frac{\text{stress}}{\text{strain}}$ the stress was calculated by multiplying the strain by $13 \cdot 7 \times 10^6$.

Stresses for specimen with $r = \frac{3}{4}$ in.

Point.	*a*	*b*	*c*	*d*	*e*	*f*
Stress in lb. per sq. in. ..	4,354 ten.	1,197 ten.	..	880 comp.	1,329 comp.	1,637 comp.

The stress diagram is shown in fig. 42. From this diagram it will be noticed that the point of zero stress is not at the centroid but at a distance x from the centroid and nearer the intrados.

The maximum or extreme fibre stresses calculated from the formulæ given are of the order of 6,600 lb. per sq. in. on the tension side, and 1,900 lb. per sq. in. on the compression side. The test results show the compressive stress at the extreme fibres to be slightly lower than the calculated stress.

The specimen with $r = 1$ in. gave a stress of 3,808 lb. per sq. in. at the point a, and the specimen with $r = 1\frac{1}{4}$ in. a stress of 2,710 lb. per sq. in. at the point a, showing that increase of radius decreases the stress at the intrados of a curved casting.

A. H. TUCKER, Government Printer, Brisbane.